THE MARINE STEAM TURBINE

J. W. SOTHERN
M.I.E.S.

THIRD EDITION

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THE MARINE STEAM TURBINE.

THE MARINE STEAM TURBINE

A Practical Description

OF THE

PARSONS MARINE TURBINE

AS PRESENTLY CONSTRUCTED, FITTED, AND RUN

INTENDED FOR THE USE OF

STUDENTS, MARINE ENGINEERS, SUPERINTENDENT ENGINEERS, DRAUGHTSMEN, WORKS' MANAGERS, FOREMEN ENGINEERS, AND OTHERS

BY

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THIRD EDITION

(*REWRITTEN UP-TO-DATE AND GREATLY ENLARGED*)

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PREFACE.

THE remarkable sea performances of the cruisers "Indomitable" and "Inflexible" and the record Atlantic passages of the "Lusitania" and "Mauretania," not to mention the phenomenal speeds (fully 36 knots) obtained in torpedo destroyer craft trials, have proved beyond question the absolute necessity for the adoption of turbine machinery where high powers and speeds are specified. As a natural result, practically all new vessels for the British Navy are to be fitted with turbines of the Parsons type, the reciprocating engine being to all intents and purposes now obsolete.

The German and other Continental engineering experts who criticised adversely the adoption of turbines for the Atlantic passenger service are now compelled to admit their mistake, and this has been recently brought home in a practical manner by means of the yearly balance-sheet of one of the great Continental steamship companies.

Foreign navies, notably those of France and Germany, are also largely adopting the turbine for cruiser and battleship design, while in America exhaustive tests are at present being carried out to compare the respective merits of the Parsons and Curtis type of turbines, which are being run against each other in vessels of identical design and displacement.

As regards coal and steam consumption, it has also been demonstrated that, under full power conditions at least, the consumption is as low and even lower than that of the best modern type reciprocating engine, while at moderate speeds the consumption is only average, and not, as often erroneously assumed, excessive.

Again, in comparing coal or steam consumptions, it is often

forgotten that a large proportion of reciprocating engines are run at a fairly high consumption, and the low figures against which the turbine results are invariably compared are in many cases the exception and not the rule in average marine practice, which comparison is obviously unfair to the turbine. Even in steamers of the tramp class the coal consumed per horse-power hour is often no less than that of a cross-channel turbine steamer, notwithstanding all that has been said to the contrary.

Just as with reciprocating engines, some turbines are much more economical than others, and it is well known by practical men that every reciprocating marine engine does not by any means work out to a consumption of, say, 1.3 lbs. of coal per I.H.P. per hour, but very often 1.8 is much nearer the figure.

The true standard of comparison should undoubtedly be the steam consumption per horse-power hour, as a pound of Welsh coal may evaporate 9 lbs. of water, while one pound of Scotch coal may only evaporate 8 lbs., or even less, so that for a given evaporation of, say, 14 lbs. steam per hour per I.H.P. the results would be 1.55 lbs. coal in the one case and 1.75 lbs. coal in the other.

It should also be remembered that the marine reciprocating engine has taken nearly a century to arrive at its present state of mechanical perfection and economical efficiency, whereas the Parsons' marine steam turbine has reached a similar degree of efficiency in the space of about a dozen years; it can therefore be safely assumed that much further improvement will yet result in turbine design and in a resultant reduced steam and coal consumption.

The recently proposed combination of turbines and dynamos, with motor drives for the shafting, would certainly considerably increase the first cost of the machinery, complicate the number and design of the working parts, and correspondingly increase the risks of breakdown, while at the same time an engine-room staff of electricians in addition to the usual engineers on watch would be necessary; the mechanical "gearing down" losses between turbine, dynamo, and motor have also to be taken into account. Whether the expected increase in turbine and propeller efficiency, due to increase of turbine revolutions and decrease of propeller revolutions, will counterbalance the foregoing disadvantages has yet to be demonstrated.

The writer trusts that the third edition of the present work will be found as interesting and useful as the previous editions, and would mention that the section on "Workshop Practice" has been specially written for the use of shop managers, draughtsmen, foremen engineers, and others engaged in turbine construction.

For kind permission to reproduce subject-matter and illustrations the writer's thanks are due to the following:—Messrs The Fore River Shipbuilding Company, Quincy, Mass., U.S.A., for descriptions and illustrations of the Curtis Marine Turbine; the proprietors of *Engineering* for illustration blocks of the turbines of R.S.Y. "Mahroussa"; the President and Council of the North-East Coast Institution of Engineers and Shipbuilders, the Institute of Naval Architects, and the American Society of Naval Engineers for extracts of papers read before those bodies; and to numerous friends and former pupils of the writer who have kindly assisted in the collection of much of the data contained in the pages which follow.

In conclusion, the writer considers he can justly claim that the volume as now presented contains the most extensive mass of reliable marine turbine data yet published.

J. W. SOTHERN.

59 BRIDGE STREET, GLASGOW,
1909.

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SECTION I.

DEFINITIONS AND GENERAL PRINCIPLES.

BEFORE commencing to describe the chief features of the Parsons turbine, it is perhaps necessary to explain clearly the meaning of certain definitions which are closely connected with the theory and practice of this type as with all other types of steam engine.

Foot-Pound.—A foot-pound is the work done in raising a weight of one pound up through a distance of one foot.

Torque.—Torque is the turning movement to which a shaft is subjected when a force is exerted to rotate the shaft against a resistance such as that of the screw propeller in water. In ordinary engines the turning effort or torque is applied by means of the crank, and in turbines by the direct energy of the steam acting on the periphery of the blade circle of the rotor.

By arranging the rotor diameter so that the peripheral velocity of the blades is equal to about half that of the steam, the maximum amount of work in foot-pounds may be extracted from each pound of steam passing through the turbine casing.

Heat.—Heat is merely a form of energy, and as such exists in two states—(1) in that of Potential or stored-up energy, and (2) in that of Kinetic or active energy. When the molecules of a body or

gas are set in rapid motion or vibration, heat is developed and work done. Consequently in the case of a steam engine, either of the reciprocating type or turbine type, the energy which produces rotation of the shaft is obtained by means of the transformation or heat energy into mechanical work.

British Thermal Unit (**B.Th.U.**).—This is taken as being equal to 778 foot-pounds of work or energy, and signifies that one heat unit, when transformed into mechanical energy, gives out 778 foot-pounds of work.

Saturated Steam.—Steam taken direct from the boilers is known as "saturated steam," as the density, or weight of water per cubic foot, is constant for any given pressure, as also is the temperature and volume. The steam supplied to all marine engines (without superheaters) is therefore of this quality, and calculations as to expansion, work done, and fall of pressure, are usually made on this assumption. The steam supplied to the H.P. turbine of a turbine engine is therefore saturated steam. Sometimes the term "dry saturated steam" is used to distinguish this quality of steam from wet steam, or steam containing water from priming.

"Wet" Steam.—If water is carried off with the steam due to priming taking place in the boilers, the steam contains more water per cubic foot than is natural to the "saturation" pressure, volume, and temperature, and it is then known as "wet steam," or "wet saturated steam."

Superheated Steam.—If saturated steam from the boilers is passed through the tubes of a superheater, the water contained in the steam is evaporated out of it, with the following results:—

1. Rise of temperature.
2. Increase of volume if pressure is kept constant; or,
3. Increase of pressure if volume is kept constant.

The chief advantage of superheated steam lies in the fact that cylinder condensation is practically eliminated, as the steam does not then readily condense when exposed to cooled surfaces: leakage is also reduced.

Another point of importance is that the specific heat of this steam being only .48 (some authorities give .5), one B.T.U. of heat supplied to the steam has the effect of raising its temperature fully two degrees, as $1 \div .48 = 2.08$.

Properties of Saturated Steam

Of from 0.5 lb. to 260 lbs. Absolute Pressure per Square Inch.

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° F.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
0.5	80.2	1105.5	1058.4	.001376	726.608
1	102.1	1112.5	1042.9	.003027	330.360
1.5	115.9	1116.7	1033.2	.004433	225.580
2	126.3	1119.9	1025.8	.005811	172.080
2.5	134.6	1122.5	1019.9	.007169	139.488
3	141.6	1124.6	1015.0	.008511	117.500
3.5	147.7	1126.4	1010.6	.009839	101.632
4	153.1	1128.1	1006.8	.01116	89.632
4.5	157.9	1129.6	1003.4	.01246	80.231
5	162.3	1130.9	1000.3	.01370	72.991
5.5	166.4	1132.1	997.4	.01505	66.428
6	170.2	1133.3	994.7	.01634	61.201
6.5	173.6	1134.3	992.3	.01762	56.761
7	176.9	1135.3	990.0	.01889	52.936
7.5	180.0	1136.3	987.8	.02016	49.610
8	182.9	1137.2	985.7	.02142	46.686
8.5	185.7	1138.0	983.8	.02268	44.097
9	188.3	1138.8	981.9	.02394	41.777
9.5	190.8	1139.5	980.1	.02547	39.261
10	193.3	1140.3	978.4	.02642	37.845
10.5	195.6	1141.0	976.7	.02767	36.145
11	197.8	1141.7	975.2	.02890	34.599
11.5	200.1	1142.4	973.6	.03026	33.045
12	202.0	1143.0	972.2	.03137	31.879
12.5	204.0	1143.6	970.8	.03260	30.678
13	205.9	1144.2	969.4	.03382	29.573
13.5	207.8	1144.8	968.1	.03504	28.536
14	209.6	1145.3	966.8	.03627	27.573
14.7	212.0	1146.1	965.2	.03797	26.360
15	213.1	1146.4	964.3	.03870	25.843
16	216.3	1147.4	962.1	.04112	24.320
17	219.6	1148.3	959.8	.04253	23.513
18	222.4	1149.2	957.7	.04594	21.766
19	225.3	1150.1	955.7	.04834	20.687
20	228.0	1150.9	953.8	.05074	19.710
21	230.6	1151.7	951.9	.05311	18.828
22	233.1	1152.5	950.2	.05549	18.022
23	235.5	1153.2	948.5	.05786	17.282
24	237.8	2153.9	946.9	.06023	16.603
25	240.1	1154.6	945.3	.06259	15.977
26	242.3	1155.3	943.7	.06495	15.401

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet
27	244.4	1155.8	942.2	.06728	14.863
28	246.4	1156.4	940.8	.05971	14.345
29	248.4	1157.1	939.4	.07196	13.896
30	250.4	1157.8	937.9	.07430	13.459
31	252.2	1158.4	936.7	.07663	13.050
32	254.1	1158.9	935.3	.07894	12.666
33	255.9	1159.5	934.0	.08128	12.300
34	257.6	1160.0	932.8	.08358	11.964
35	259.3	1160.5	931.6	.08590	11.640
36	260.9	1161.0	930.5	.08821	11.337
37	262.6	1161.5	929.3	.09050	11.050
38	264.2	1162.0	928.2	.09282	10.773
39	265.8	1162.5	927.1	.09510	10.515
40	267.3	1162.9	926.0	.09740	10.267
41	268.7	1163.4	924.9	.09946	10.054
42	270.2	1163.8	923.9	.1020	9.806
43	271.6	1164.2	922.9	.1042	9.592
44	273.0	1164.6	921.9	.1065	9.386
45	274.4	1165.1	920.9	.1088	9.191
46	275.8	1165.5	919.9	.1111	9.003
47	277.1	1165.9	919.0	.1134	8.821
48	278.4	1166.3	918.1	.1156	8.650
49	279.7	1166.7	917.2	.1179	8.482
50	281.0	1167.1	916.3	.1202	8.322
51	282.3	1167.5	915.4	.1224	8.170
52	283.5	1167.9	914.5	.1247	8.021
53	284.7	1168.3	913.6	.1269	7.880
54	285.9	1168.6	912.8	.1292	7.741
55	287.1	1169.0	912.0	.1314	7.610
56	288.2	1169.3	911.2	.1337	7.482
57	289.3	1169.7	910.4	.1357	7.370
58	290.4	1170.0	909.6	.1382	7.238
59	291.6	1170.4	908.8	.1404	7.123
60	292.7	1170.7	908.0	.1426	7.011
61	293.8	1171.1	907.2	.1449	6.902
62	294.8	1171.4	906.4	.1471	6.798
63	295.9	1171.7	905.6	.1493	6.696
64	296.9	1172.0	904.9	.1516	6.596
65	298.0	1172.3	904.2	.1538	6.502
66	299.0	1172.6	903.5	.1560	6.410
67	300.0	1172.9	902.8	.1583	6.318
68	300.9	1173.2	902.1	.1604	6.233
69	301.9	1173.5	901.4	.1627	6.147

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density of Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	L s.	Cubic Feet.
70	302.9	1173.8	900.8	.1650	6.059
71	303.9	1174.1	900.3	.1671	5.984
72	304.8	1174.3	899.6	.1693	5.905
73	305.7	1174.6	898.9	.1716	5.829
74	306.6	1174.9	898.2	.1738	5.764
75	307.5	1175.2	897.5	.1760	5.683
76	308.4	1175.4	896.8	.1782	5.610
77	309.3	1175.7	896.1	.1803	5.544
78	310.2	1176.0	895.5	.1826	5.476
79	311.1	1176.3	894.9	.1848	5.411
80	312.0	1176.5	894.3	.1870	5.348
81	312.8	1176.8	893.7	.1892	5.286
82	313.6	1177.1	893.1	.1912	5.230
83	314.5	1177.4	892.5	.1936	5.167
84	315.3	1177.6	892.0	.1957	5.109
85	316.1	1177.9	891.4	.1980	5.052
86	316.9	1178.1	890.8	.2001	4.996
87	317.8	1178.4	890.2	.2023	4.942
88	318.6	1178.6	889.6	.2046	4.889
89	319.4	1178.9	889.0	.2067	4.837
90	320.2	1179.1	888.5	.2088	4.790
91	321.0	1179.3	887.9	.2111	4.737
92	321.7	1179.5	887.3	.2133	4.688
93	322.5	1179.8	886.8	.2154	4.642
94	323.3	1180.0	886.3	.2176	4.595
95	324.1	1180.3	885.8	.2198	4.549
96	324.8	1180.5	885.2	.2220	4.505
97	325.6	1180.8	884.6	.2241	4.462
98	326.3	1181.0	884.1	.2263	4.419
99	327.1	1181.2	883.6	.2286	4.375
100	327.9	1181.4	883.1	.2307	4.335
101	328.5	1181.6	882.6	.2329	4.305
102	329.1	1181.8	882.1	.2350	4.256
103	329.9	1182.0	881.6	.2372	4.216
104	330.6	1182.2	881.1	.2393	4.178
105	331.3	1182.4	880.7	.2415	4.140
106	331.9	1182.6	880.2	.2437	4.104
107	332.6	1182.8	879.7	.2458	4.068
108	333.3	1183.0	879.2	.2480	4.033
109	334.0	1183.3	878.7	.2502	3.998

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
110	334.6	1183.5	878.3	.2523	3.963
111	335.3	1183.7	877.8	.2545	3.930
112	336.0	1183.9	877.3	.2566	3.897
113	336.7	1184.1	876.8	.2588	3.865
114	337.4	1184.3	876.3	.2610	3.832
115	338.0	1184.5	875.9	.2631	3.801
116	338.6	1184.7	875.5	.2653	3.770
117	339.3	1184.9	875.0	.2674	3.740
118	339.9	1185.1	874.5	.2696	3.710
119	340.5	1185.3	874.1	.2717	3.681
120	341.1	1185.4	873.7	.2738	3.652
121	341.8	1185.6	873.2	.2760	3.623
122	342.4	1185.8	872.8	.2781	3.595
123	343.0	1186.0	872.3	.2803	3.567
124	343.6	1186.2	871.9	.2824	3.541
125	344.2	1186.4	871.5	.2846	3.514
126	344.8	1186.6	871.1	.2867	3.488
127	345.4	1186.8	870.7	.2889	3.462
128	346.0	1186.9	870.2	.2910	3.436
129	346.6	1187.1	869.8	.2931	3.411
130	347.2	1187.3	869.4	.2951	3.388
131	347.8	1187.5	869.0	.2974	3.362
132	348.3	1187.6	868.6	.2996	3.338
133	348.9	1187.8	868.2	.3017	3.315
134	349.5	1188.0	867.8	.3038	3.291
135	350.1	1188.2	867.4	.3060	3.268
136	350.6	1188.3	867.0	.3080	3.246
137	351.2	1188.5	866.6	.3102	3.224
138	351.8	1188.7	866.2	.3123	3.201
139	352.4	1188.9	865.8	.3145	3.180
140	352.9	1189.0	865.4	.3166	3.159
141	353.5	1189.2	865.0	.3187	3.138
142	354.0	1189.4	864.6	.3209	3.117
143	354.5	1189.6	864.2	.3230	3.096
144	355.0	1189.7	863.9	.3251	3.076
145	355.6	1189.9	863.5	.3272	3.056
146	356.1	1190.0	863.1	.3293	3.037
147	356.7	1190.2	862.7	.3315	3.017
148	357.2	1190.3	862.3	.3336	2.998
149	357.8	1190.5	861.9	.3357	2.979
150	358.3	1190.7	861.5	.3378	2.960
151	359.0	1190.9	861.1	.3400	2.941

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
152	359.5	1191.0	860.7	.3421	2.923
153	360.0	1191.2	860.4	.3442	2.905
154	360.5	1191.4	860.0	.3463	2.887
155	361.1	1191.5	859.6	.3484	2.870
156	361.6	1191.7	859.2	.3505	2.853
157	362.1	1191.8	858.9	.3527	2.836
158	362.6	1192.0	858.5	.3548	2.818
159	363.1	1192.1	858.1	.3569	2.802
160	363.6	1192.3	857.8	.3590	2.785
165	366.0	1192.9	856.2	.3696	2.706
170	368.2	1193.7	854.5	.3801	2.631
175	370.8	1194.4	852.9	.3905	2.559
180	372.9	1195.1	851.3	.4011	2.493
185	375.3	1195.8	849.6	.4115	2.430
190	377.5	1196.5	848.0	.4220	2.370
195	379.7	1197.2	846.5	.4324	2.313
200	381.7	1197.8	845.0	.4419	2.263
210	286.0	1199.1	841.9	.463	2.157
220	389.9	1200.3	839.2	.484	2.065

Adiabatic Expansion.—If steam expands in a cylinder or turbine casing, and neither receives heat from any external source nor gives out any heat externally, then the expansion is said to be "adiabatic," and all work done in the cylinder or turbine is obtained at the expense of the internal heat of the steam, which in falling in pressure and temperature conforms to this condition, and part of which condenses. In the cylinders of a marine engine of the reciprocating type, the expansion is approximately hyperbolic or isothermal, and in a turbine the expansion is approximately "adiabatic."

Hyperbolic or Isothermal Expansion.—This is founded on the well-known law of Boyle and Marriot that the pressure of a gas varies inversely as the volume; or, as it is expressed—

Rule, $P_1 \times v_1 = p_2 \times V_2 = \text{Constant.}$

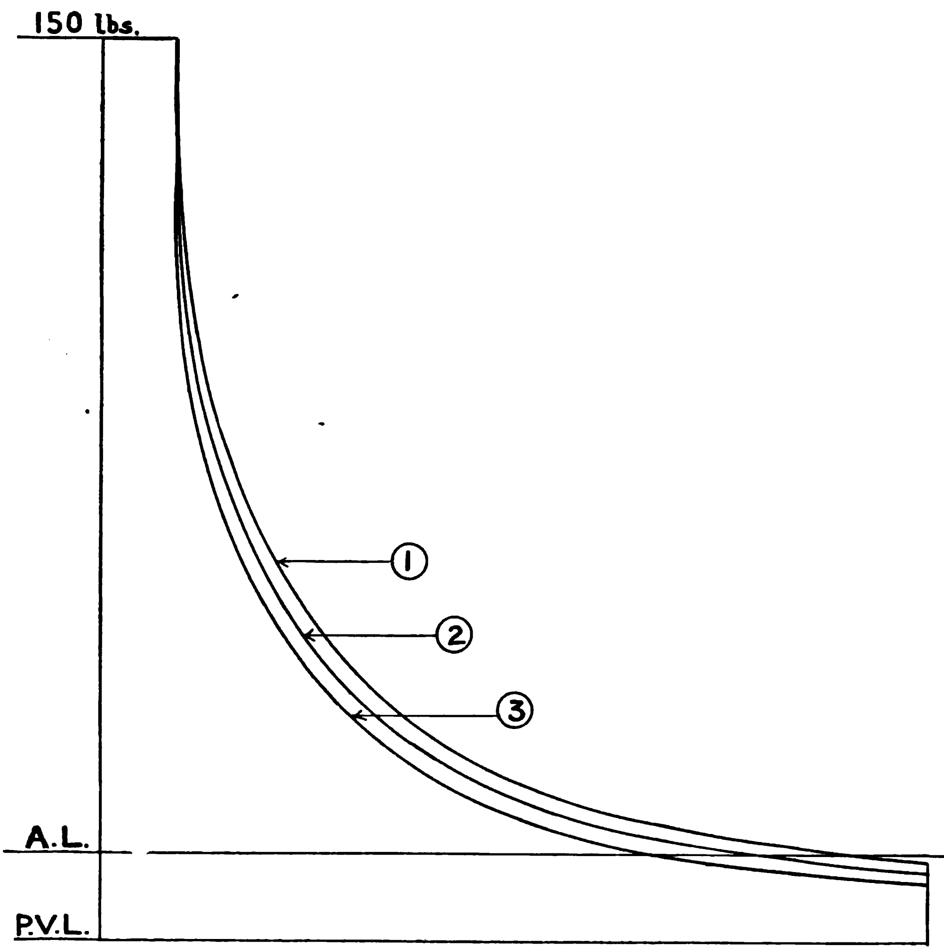
Where P_1 = Initial pressure. p_2 = Final pressure.
v_1 = ,, volume. V_2 = ,, volume.

$$\text{Therefore, } \frac{P_1 \times v_1}{p_2} = V_2\,;$$

$$\text{and, } \frac{P_1 \times v_1}{v_2} = p_2\,;$$

$$\text{or, } \frac{p_2 \times V_2}{P_1} = v_1\,;$$

$$\text{and, } \frac{P_2 \times V_2}{v_1} = P_1\,;$$



Expansion Curves of Steam.

(1) Isothermal or Hyperbolic Curve, $P \times V =$ Constant (Perfect Gas).
(2) Saturation Curve, $P \times V^{1\frac{1}{16}} =$ Constant (Reciprocating Engine, approximately).
(3) Adiabatic Curve, $P \times V^{\frac{10}{9}} =$ Constant (Turbine Engine, approximately).

As explained before, ordinary steam, being an imperfect gas, does not exactly follow out the above law, but deviates in the direction of the "saturated steam" curve, as shown in the diagrams, in the case of reciprocating engine cylinders.

Dryness Fraction (or Factor).—In considering the actual work done by steam between the rows of blades of a turbine, it is important that the dryness fraction be taken into account, as the result greatly depends on this quantity. After work is done by adiabatic expansion, the steam contains a certain amount of water, which proportionally reduces the internal heat still left in the steam. The dryness fraction is the ratio between the weight of dry steam per pound and the weight of the dry steam and water added together;

$$\text{Or, } \frac{\text{Weight of dry steam}}{\text{Weight of dry steam} + \text{weight of water}} = \text{Dryness Fraction.}$$

Suppose the water to be 25 per cent. of each pound weight of mixture,

$$\text{Then, } \frac{100 - 25}{100} = \frac{75}{100} = \frac{15}{20} = \frac{3}{4} = \text{Dryness Fraction (or Factor).}$$

So that after expansion and work done by the steam the actual units or foot-pounds of energy left are equal to the internal heat units multiplied by the fraction $\frac{3}{4}$.

Total Heat of Steam.—By the total heat of saturated, or boiler steam, is meant the number of heat units required to produce one pound of steam from a temperature of 32° Fahr. to any given temperature and pressure. The total heat includes the latent heat of steam formation and the sensible or thermometer heat.

RULE. $1083 + .3 \times T^\circ = \text{Total Heat (above } 32^\circ \text{ Fahr.)}.$

Where T° = Temperature of the steam (Fahr.).

Internal Heat of Steam.—By this is meant the heat or energy required to change one pound of water into steam at any given pressure.

External Heat of Steam.—By this is meant the heat required to produce increase of volume (water to steam) against an external resistance or pressure.

Latent Heat of Steam.—The sum of the Internal heat and External heat is equal to the latent heat.

The Latent Heat can be calculated as follows:—

RULE. $1114 - .7 \times T^\circ = \text{Latent Heat}.$

Where T° = Temperature of the steam (Fahr.).

EXAMPLE.—Calculate the Total Heat, Latent Heat, and Sensible Heat of 1 lb. of steam at 160 lbs. pressure by gauge.

$160 + 15 = 175$ lbs. absolute pressure and 371° Temperature (from Table, page 3),

Then, $1083 + .3 \times 371 = 1194.3$ Total Heat,

and $1114 - .7 \times 371 = 854.3$ Latent Heat.

Therefore $371^\circ - 32^\circ = 333.9$ Sensible Heat.

NOTE.—The above are all calculated from a temperature of 32° Fahr.

Potential Energy is the energy contained or stored up in steam of a given pressure and temperature, the amount of energy contained increasing with the pressure and the temperature.

Kinetic Energy is the result of setting free the potential or stored-up energy of the steam, which then shows as active energy in the performance of work. In a steam engine the steam acts on the pistons, and by causing motion to take place work is done, and, as a result, the steam falls in pressure and in temperature. In a turbine, the steam at a given pressure and velocity leaves the first row of guide blades, and striking the first row of moving blades gives up part of its kinetic energy, which results in a decrease in pressure and in heat. It then enters the next row of guide and moving blades, where more energy is given up, and a further decrease in pressure and in heat takes place. This is repeated row after row, the steam falling in pressure and in temperature, but, be it noted, increasing in volume. This increase of volume would produce increase of velocity if the blades were not made (1) longer, or (2) spaced farther apart. Both methods, separately and combined, are adopted at different expansion stages of the Parsons turbine, as will be shown later.

The effective kinetic energy of each pound of steam supplied to the turbine is employed in exerting a torsional stress on the shaft, and thus produces rotation. In the boilers the potential energy of the steam is generated, and in the cylinders of an ordinary engine, or in the turbine casing of a turbine engine, the potential energy is liberated and transformed into kinetic energy, and a certain number of foot-pounds of work are done in causing rotation of the shaft. In all steam engines only a very small portion of the total heat of the steam can be changed into foot-pounds of useful work, seldom more than about 14.3 per cent., or $\frac{1}{7}$, as shown by the following:—

Suppose the consumption of coal is 1.5 lb. per I.H.P. per hour—

$$\text{Then, } \frac{60 \times 33000}{1.5 \times 11500 \times 778} = \frac{1}{7}, \text{ or } 14.3 \text{ per cent.}$$

NOTE.—Allowing 11500 units of heat to be given up per pound of coal.

This applies equally to a turbine or reciprocating engine. Suppose, then, that a pound of steam in passing through a turbine gives up, say, 200 units of heat,

Then, $200 \times 778 = 155600$ foot-pounds of energy given out.

This means that 155600 foot-pounds of work have been done in rotating the shaft, neglecting blade leakage, blade friction, and other losses.

If, then, we know that a certain number of pounds of steam are used in a given time, say, for example, 1000 lbs. per minute, then,

$$1000 \times 155600 = 155600000 \text{ foot-pounds of energy developed.}$$

After losses are allowed for, the actual foot-pounds of work done, or kinetic energy expended, is exactly equal to the units of heat effectively applied, multiplied by 778.

Blade Friction.—It is important that the surfaces of the blades should be as smooth as possible, since the friction produced by the flow of steam across the blades somewhat seriously affects the efficiency of the turbine. It will also be obvious that water in the steam will produce a similar result, so that one clear advantage of superheating the steam is the reduction in frictional resistance set up by saturated steam.

Expansion of Steam.—In a modern marine triple-expansion engine with cylinder areas of H.P. to L.P. as 1 is to 7.5, and with a "linked up" cut-off in the H.P. of $\frac{1}{3}$ stroke, the total number of expansions of steam would be 22.5, as $7.5 \times 3 = 22.5$ (by volumes).

In the turbine engine, however, the number of expansions of steam is much more than this, from 125 to 140 expansions being readily obtained. With an H.P. turbine initial pressure of 150 lbs. absolute, and a condenser vacuum of 29 in., or back pressure of say 1 lb., the steam would expand about 150 times by pressures, as $150 \div 1 = 150$. From this it will be seen that more work can be got out of the steam by the greatly increased number of steam expansions, and the importance of obtaining a very high vacuum in the condensers will be obvious.

Condensation.—In all ordinary engine cylinders losses from condensation are more or less serious, the cause being, as is well known, the difference of temperature existing between the initial and exhaust pressures, alternately heating up and cooling down the cylinder walls. This results in a certain amount of steam condensing into water without doing work.

In a turbine casing no such variation in temperature exists, as the range of temperature is practically constant throughout the turbine from end to end, the steam entering at one end at a high temperature, and flowing to the other end continuously as it falls gradually in pressure and temperature.

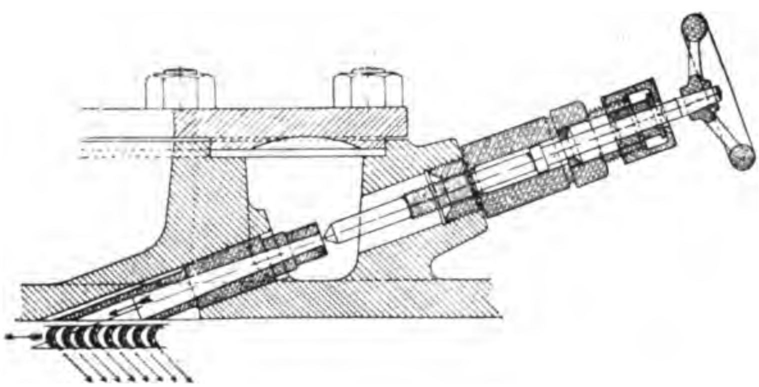
Again, if condensation does take place, the water formed is not so troublesome to get rid of as in an ordinary cylinder, as it simply drains away to the exhaust end of the turbine and so to the air pumps.

The condensation which does occur in the turbines is that due to the adiabatic expansion of the steam during the transformation from potential to kinetic energy, as described elsewhere.

Principle of Turbine.—The steam turbine is a machine designed to convert the kinetic energy of steam into direct rotary motion. The two principal types of turbine are—(1) Impulse Turbines, those arranged with expanding nozzles in which the high velocity of discharge impinges against a series of small buckets secured on the circumference of a large wheel keyed to the driving shaft, the De-Laval turbine being an example of this type; and those (2) Impulse-reaction Turbines, in which the steam passes through a number of rings of fixed blades and of moving blades, expanding as it travels, an example of which is found in the Parsons turbine.

Work by impulse is produced by high velocities, and as the work is done at the expense of the internal heat, water is formed, which thus diminishes the heat left.

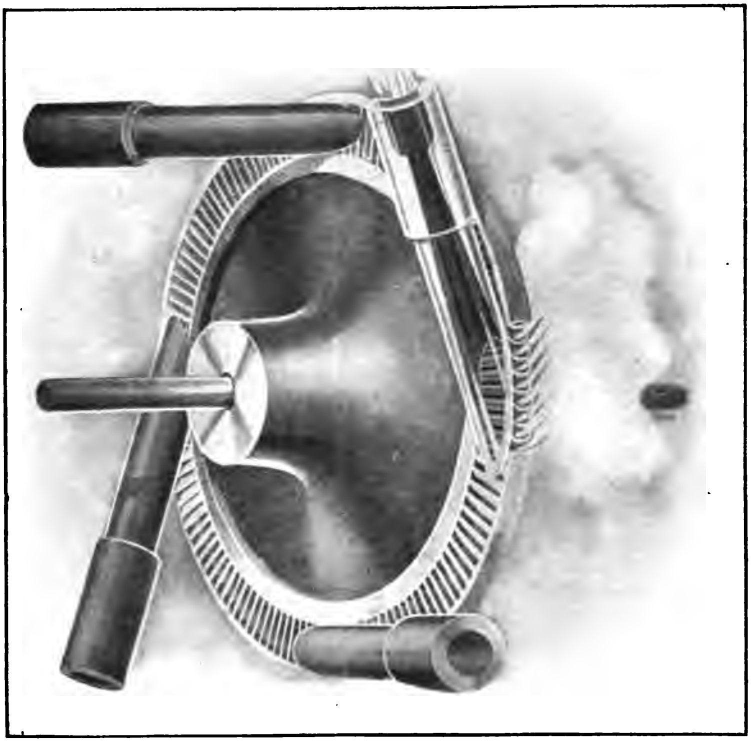
The Parsons turbine is generally called a reaction turbine, although the correct term should be "impulse-reaction" turbine, as the steam actually does act first by impulse from the guide to the moving blades and afterwards works by reaction from the moving to the guide blades.



ARRANGEMENT OF NOZZLE AND SHUTTING-OFF VALVE.
De-Laval Turbine.

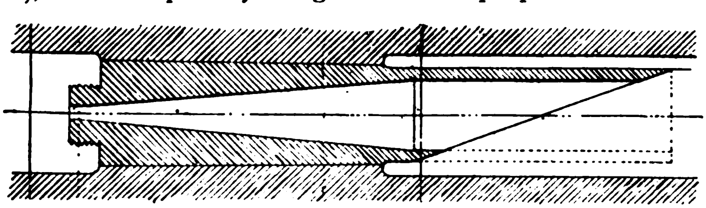
De-Laval Turbine.—In the specially shaped diverging nozzle of the De-Laval turbine shown in the sketch, the steam expands down to the required exhaust pressure, and the resultant kinetic energy acquired is applied direct to the small buckets or vanes, the steam being in consequence at a very high velocity. To obtain the best efficiency the circumferential velocity of the turbine blades should be equal to about half the velocity of the steam, and this, of course, demands a very high revolution speed. In the De-Laval turbine the speed is often as high as 20,000 revolutions per minute; this can, however, be reduced by suitable gearing to about 2,000 revolutions per minute, but as even this is too high for the shafting of marine engines, the non-adaptability of this turbine for marine purposes will be obvious. The steam is admitted to the nozzles (usually four or six in number) and controlled by regulating hand valves.

It is worthy of notice that in this type of turbine the turbine wheel is rotated by steam at the expanded or lowest pressure, as the actual



View of De-Laval Turbine in Action.

expansion takes place in the nozzle (and not within the vanes or buckets), which is specially designed for that purpose.



Nozzle of De-Laval Turbine.

The De-Laval type of turbine is much in use for the driving of dynamos, and many steamers are supplied with this turbine for the lighting set of the ship.

Parsons Turbine.—In this, the latest and most successful development of marine engineering, steam is admitted direct from the boilers to blades on the shaft drum, thus doing away with the necessity for piston valves or slide valves, cylinders, pistons, piston rods, crossheads, connecting rods, cranks, eccentrics, eccentric rods, and links, &c.

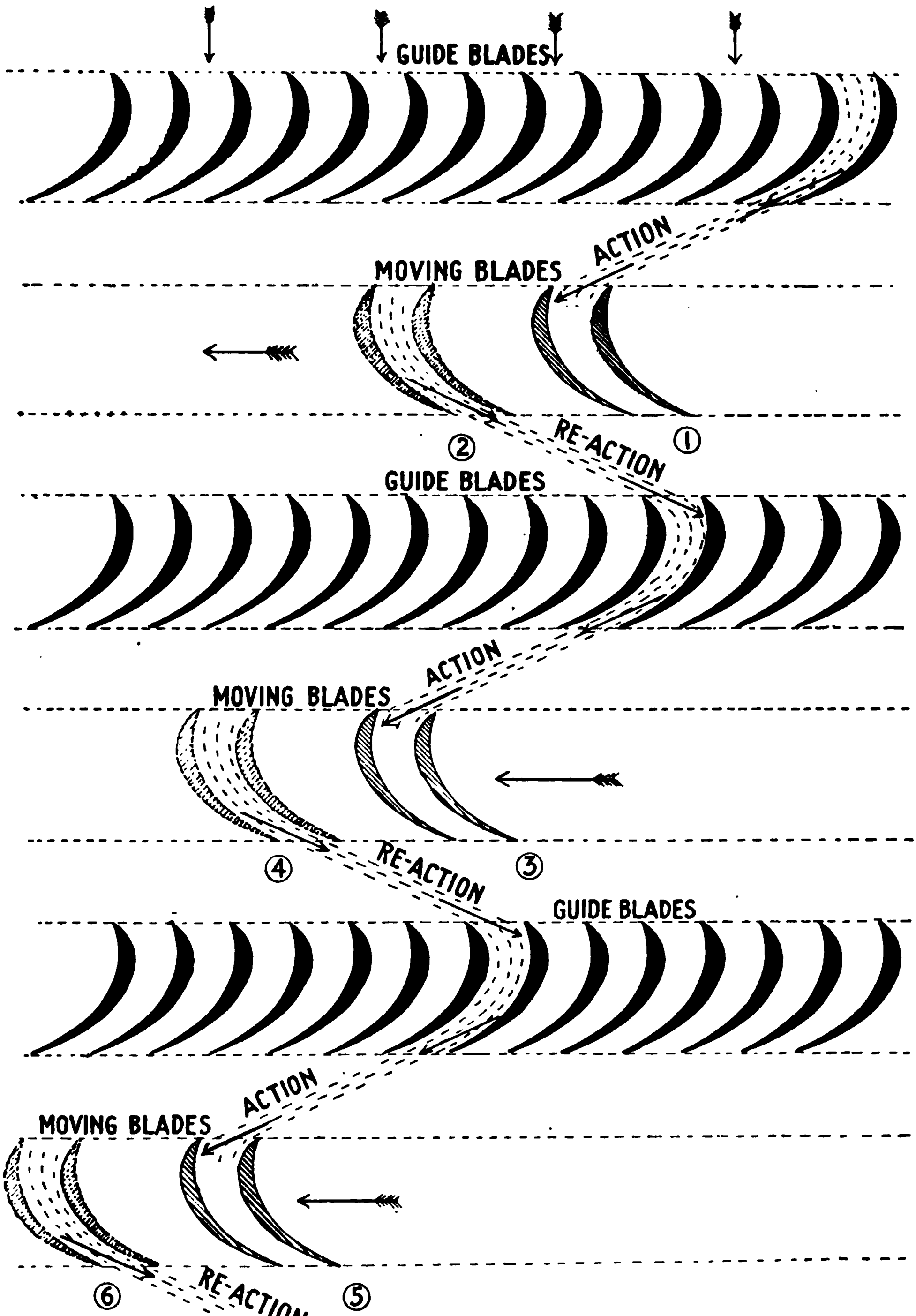
The power to rotate the shaft is therefore applied direct, which in itself constitutes one of the conditions of an ideal engine. The inventor, the Hon. C. A. Parsons, M.A., F.R.S., gives the following brief description of the turbine :—

"The Parsons turbine consists of a cylindrical case with numerous rings of inwardly projecting blades. Within this cylinder, which is of variable internal diameter, is a shaft or spindle, and on this spindle are mounted blades, projecting outwardly, by means of which the shaft is rotated. The former are called fixed or guide blades, and the latter revolving or moving blades. The diameter of the spindle is less than the internal diameter of the cylinder, and thus an annular space is left between the two. This space is occupied by the blades, and it is through these the steam flows. The steam enters the cylinder by means of an annular port at the forward end ; it meets a ring of fixed guide blades which deflect it so that it strikes the adjoining ring of moving blades at such an angle that it exerts on them a rotary impulse. When the steam leaves these blades it has naturally been deflected. The second ring of fixed blades is therefore interposed, and these direct the steam on to the second ring of rotating blades. The same thing occurs with succeeding rings of guide and moving blades until the steam escapes at the exhaust passage."

Steam from the boiler is admitted by suitable hand valves to the forward end of the casing surrounding the blades, and after passing through a ring of guide blades fixed to the casing, strikes the first ring of shaft or rotor blades ; it next passes through the second ring of fixed blades, then the second ring of rotor blades, and so on, passing alternately ring after ring of guide and rotor blades, and so rotating the shaft, until it finally exhausts at the other end of the turbine casing at a reduced pressure.

Parallel Flow.—Parsons' marine turbine is known as that of the impulse and reaction "parallel flow" type, as the steam enters the guide vanes in lines parallel more or less to the shaft axis, and in this way passes from end to end of the turbine, reacting, expanding, and falling in pressure as it travels.

Action of Steam.—As will be seen from the foregoing, the steam striking the blades imparts a turning movement to the shaft, and after reacting and passing through the series of rings of vanes of the H.P. turbine exhausts simultaneously into the two L.P. turbines, one on either side, and expanding through the longer casing and shaft blades of these turbines, finally exhausts, at a low absolute pressure of from 1½ to 2 lbs., into the condensers, one for each L.P. turbine.



Path traced out by Steam in Parsons Turbine.

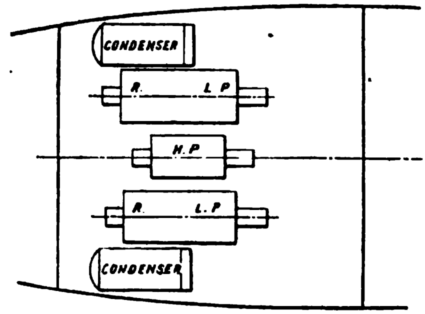
Flow of Steam through Blades.—The diagram on page 15 shows graphically the path followed out by the steam as it passes through each successive ring of fixed and moving blades. Observe that the steam, after passing through the first ring of guide blades, strikes the first ring of rotor or moving blades and by the action set up assists in rotating the shaft; by the time the steam has changed its direction the rotor has moved round a certain distance (from 1 to 2), and the reaction of the steam, due to its somewhat sudden change of direction, still further assists in rotating the shaft. The steam then leaves the rotor blades and enters the next ring of guide blades, where, after again being deflected in its path, it enters the next ring of moving blades, where the action and reaction process is again repeated; leaving the second ring of moving blades at position 4 the steam enters the third ring of guide blades at a point 5 still farther round the circumference, and so on for each of the following rings.

It should be noted that the steam leaving the moving blades is deflected by the blade curvature, and strikes the casing blades, which, if free to revolve, would be acted on by the steam and moved round similarly to the rotor blades, but in the opposite direction; instead of this taking place, however, the casing blades being fixed resist the impact, and the resulting reaction throws back the steam, the velocity of which is thus increased. The pressure is therefore utilised in augmenting the steam speed, hence the statement that "in the guide blades the steam does work on itself to increase its own velocity" The steam thus describes a somewhat zigzag path in passing along the rotor, its direction being not unlike that of a screw thread. Work is done at each ring of blades and heat given up, expansion of the steam taking place in due proportion, so that the velocity of flow increases, and to allow for this the lengths and spacings of the blades must be increased to maintain the same ratio between the blade velocity and the steam velocity, upon which the turbine efficiency depends.

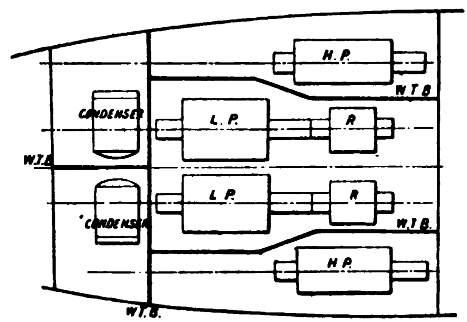
The diagram shows the imaginary path described by a small portion of steam, and the dotted blades show the circumferential advance of the rotor blades at each ring, which produces the thread-like path traced out by the steam.

Turbine Arrangements.—In steamers of ordinary size for either channel or deep-sea service, the standard arrangement consists of five turbines, three for ahead and two for reverse running; three shafts are fitted with one propeller on each, the reverse turbines being placed within the L.P. turbine casings aft. In exceptionally large steamers, such as the "Lusitania" and "Mauretania," four lines of shafting are arranged, with two ahead H.P. turbines and two ahead L.P. turbines, also two independent reverse turbines on the inner shafts. This arrangement, with the further addition of other two reverse turbines and two cruising turbines, is carried out in the case of large battleships and cruisers; sometimes the cruising turbines are compound,

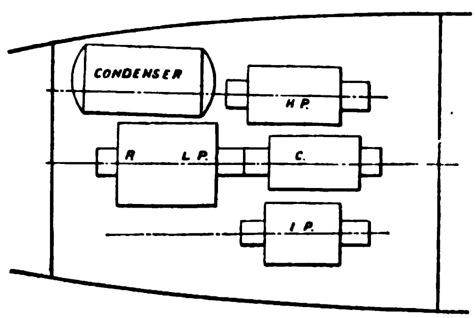
(1) **Standard arrangement,** one H.P. and two L.P. turbines, three shafts, one propeller to each.



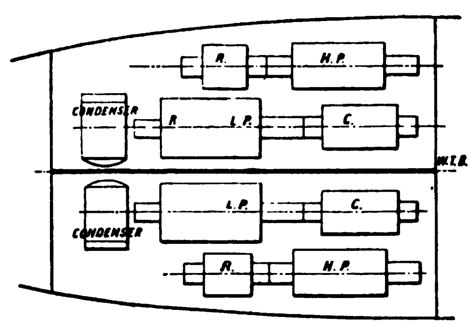
(2) **Large passenger steamer arrangement,** two H.P., two L.P., and two independent reverse turbines, four shafts, one propeller to each.



(3) **Torpedo craft arrangement,** one cruising, one I.P., one H.P., and one L.P. turbine, three shafts, one propeller to each.



(4) **Battleship or cruiser arrangement,** two cruising ahead, two H.P. ahead, two L.P. ahead turbines, also two H.P. reverse, and two L.P. reverse turbines, four shafts, one propeller to each.



one H.P. and one M.P., but generally both are of the same size, and receive direct steam from the boilers simultaneously. It should be noted that the Admiralty have decided to discard cruising turbines altogether in future, as in most cases the consumption at the low powers developed by these turbines does not justify their existence, in addition to the loss of power produced by the turbine blade resistance when running ahead or astern with the main turbines. Cruisers of the "Inflexible"-"Indomitable" type have ten turbines fitted, four ahead turbines—two H.P. and two L.P.—and four reverse—two H.P. and two L.P.—also two cruising turbines fitted, one on each H.P. turbine shaft, and intended for low cruising speeds and powers.

In torpedo craft the three-shaft arrangement is often carried out, but the turbines are arranged in triple series, one H.P. (centre), one M.P. (wing), and one L.P. (wing). Sometimes cruising turbines are fitted in addition to these in the case of large high-speed destroyers. The foregoing are the arrangements of turbines in present practice, but other arrangements have been proposed by the Parsons Company.

As regards the new combination arrangement of reciprocating engines and turbine, the steamers at present under construction are fitted with two wing triple or quadruple engines, both exhausting at a pressure below the atmosphere into the turbine on the centre shaft. An alternative design consists of one centre reciprocating engine exhausting into two wing turbines.

Steam Flow through Turbines.—In the standard turbine arrangement of five turbines—three ahead and two reverse—the steam, after expanding through the H.P. turbine, exhausts to both L.P. turbines simultaneously, and then to the two condensers. In the "Lusitania" design, the steam expands through each H.P. turbine, then through each L.P. turbine to the condensers of each respective side.

In the "Inflexible"-"Indomitable" class turbine arrangement, at full ahead power, the steam expands through each H.P., then each L.P., and then to the condensers of each side. At reduced ahead power, the steam first expands through each cruising turbine, then through each H.P. and L.P. turbine of each side, finally exhausting to the condensers. In running astern the steam first expands through the H.P. reverse turbine, then the L.P. reverse turbine, and finally exhausts to the condenser.

In the destroyer triple arrangement at full power, the steam first expands through H.P. turbine, then M.P. turbine, and L.P. turbine to the condenser, and at reduced power or cruising speed, the steam first expands through the cruising turbine, H.P. turbine, M.P. turbine, and L.P. turbine to the condenser.

Increase of Steam Volume.—To allow of the steam increasing in volume, as fall of pressure takes place, the various sets of blades increase in length from the forward to the after end, the clearance spaces between the blades also increasing in proportion, which necessitates packing pieces of a larger size being employed. The

blades also vary in shape or curvature, being flatter in section aft than forward. Each set of blades for each expansion requires its own allowance for expansion of metals by heat, so that the working clearance between the blades and casings or drums increases slightly throughout the turbine from forward aft. One of the practical difficulties met with in turbine construction at present is the correct adjustment for this expansion, as slight mishaps have occurred in one or two instances owing to fouling of the parts when heated up, the clearance allowance being insufficient.

Strictly, each successive ring of blades should be either of a wider pitch or greater height than the preceding one, as the steam is continuously falling in pressure and expanding in volume, but the Hon. C. A. Parsons considers the present arrangement quite near enough for practical purposes for all the difference that results.

NOTE.—The various extracts which follow, are from a paper read at the Forty-ninth Meeting of the Institution of Engineers and Shipbuilders in Scotland, on 24th October 1905, and entitled "The Determination of the Principal Dimensions of the Steam Turbine, with Special Reference to Marine Work," by Mr E. M. Speakman, Associate Member, &c., and are reprinted here by kind permission of the Council of that Institution. Describing the action of steam in passing through the successive rings of blades, &c., Mr E. M. Speakman says:—

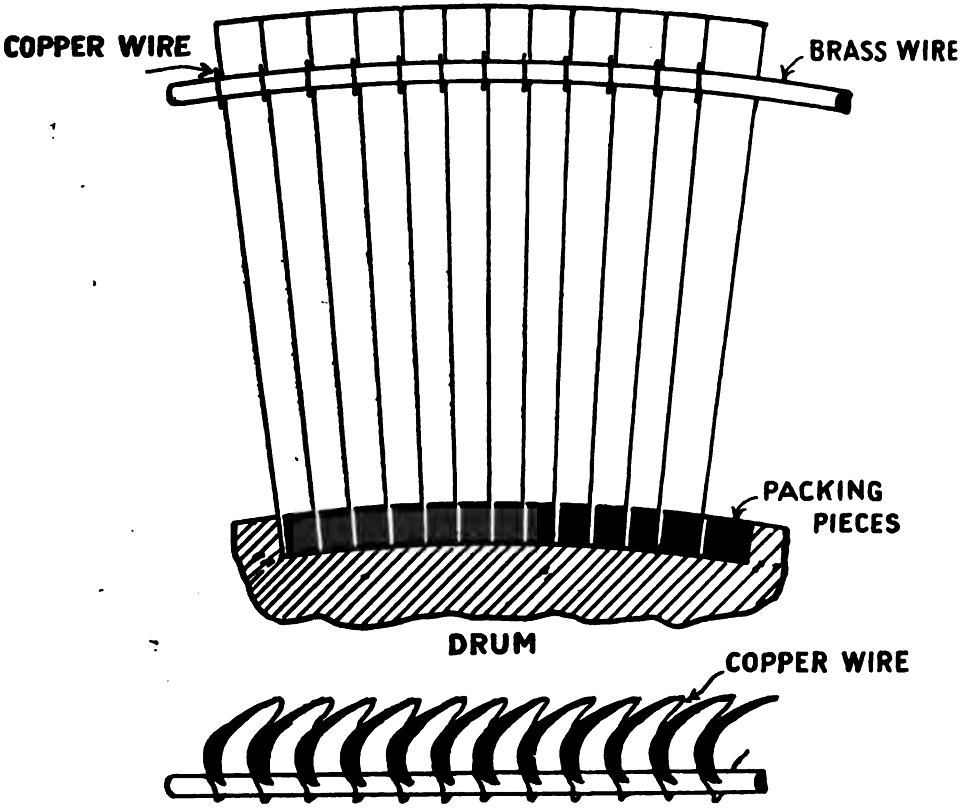
"Restricting attention to the design of the Parsons type of turbine, a few notes on the action of the steam among the blades may be of interest. Expanding through a definite range of temperature and pressure, steam exerts the same energy, whether it issues from a suitable orifice or expands against a receding piston. Two transformations of energy take place in the steam turbine—first, from thermal to kinetic energy; secondly, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine, the radical difference between the two lying in the low density of steam compared with water, and the wide variation of its volume under different temperatures and pressures.

"Fig. 5 gives a sectional elevation of a marine-turbine blading arrangement, and though this is only for an H.P. cylinder the principle is exactly the same throughout. The expansion, which is approximately adiabatic, is carried out in this annular chamber from A to B, which essentially resembles a simple divergent steam nozzle, but with this difference, that whereas in a nozzle the heat energy of the working steam is expended upon itself in producing high velocities, in Parsons' turbine the total expansion is subdivided into a number of steps, in each of which a certain dynamic relationship between jet and vane is maintained. The expansion of steam at any one stage is typical of its working throughout the turbine. Each stage consists of a ring of stationary blades which give direction and velocity to the steam, and a ring of moving blades that immediately convert the energy of velocity into useful torque. The total torque on the shaft is due to the impulse of steam entering the moving blades and to reaction as it leaves them, this process being repeated throughout the turbine.

"Leakage past the revolving portion of the spindle at D is almost entirely prevented by the ingenious form of frictionless packing, shown on a larger

scale on Fig. 6. The fine clearances and the sudden increase of section have the effect of alternately wire-drawing and expanding the steam, so that at successive grooves it becomes increasingly difficult for the steam to leak past the fine clearances. In the astern turbines, a radial form of packing, depending on fine tip clearances, must be adopted owing to the difference in expansion between spindle and cylinder. Numerous varieties of these forms of packing exist, some of them being extremely efficient in their action.

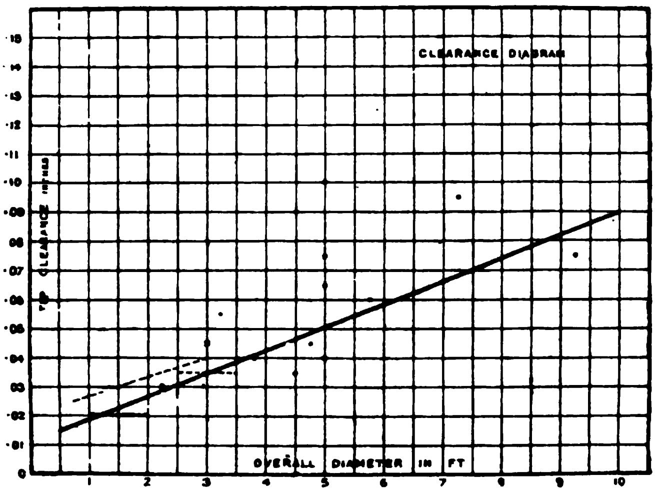
"The laws governing the best theoretical velocity of steam and blades are



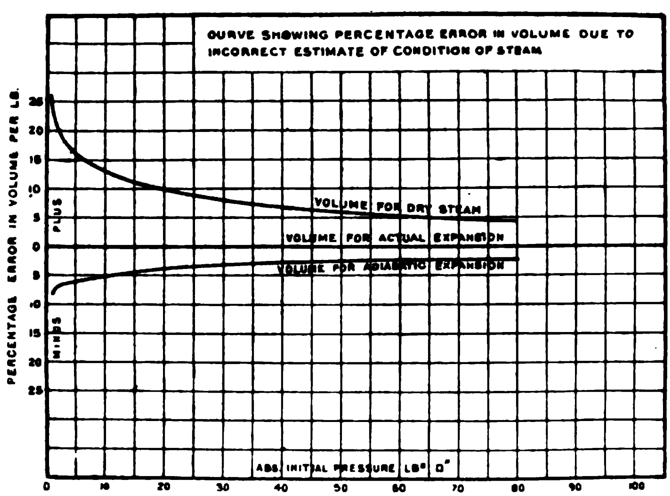
Elevation and Plan of Rotor Blades in Position, showing how secured (full size).

similar to those for water turbines, but in practice some modification is necessary, and the best ratio of blade speed and steam speed is still a matter of opinion. The ideal condition for impulse turbines occurs when the peripheral velocity of the buckets is one-half that of the jet, or in reaction turbines, when it is equal to it.

"Parsons' turbines, however, have been built with V_t, Fig. 7, varying from .25 to .85 of V_s, where V_t represents blade velocity at mean diameter, and V_s the steam speed due to expansion across the row in question. A very usual



Tip Clearance Diagram.



Correction Factors for Condition of Steam.

ratio in electrical work for large units has been $\frac{V_t}{V_s} = 0.6$, but this involves a greater number of rows than is possible in marine work, and the ratio must be reduced. These ratios need very careful calculation. The steam consumption must be accurately known in order to proportion them correctly throughout the turbine, and the necessity (which is inevitable with the present form of caulking piece) of having the same area of openings in so many rows while the steam volume increases so rapidly that it adds to the difficulty of close calculation.

"The potential energy of the steam, corresponding to the 'head' in water turbines, can easily be calculated for given pressure differences.

$$\text{B.Th.U.} \times 778 = \text{Energy in foot-pounds per pound of steam} = e.$$

$$V_s = 8\sqrt{e} = 223\sqrt{\text{B.Th.U.}}\ *$$

"For a given blade velocity, it is obvious, then, that the speed ratio between jet and vane must affect the number of stages, and the greater the ratio of V_t to V_s the greater will be the required number of rows, that is, to obtain the required V_s at each stage a smaller pressure drop per row is necessary, or *vice versa*.

"The best blading arrangement, scientifically and commercially, is the result of much theory and practice. The mean diameter is an arbitrary dimension capable of wide variation without affecting the efficiency, provided that the number of rows is correct; it is found by assuming, from experience, a blade velocity, whence—

$$\text{Mean diameter in inches} = \frac{\text{Blade velocity in feet per sec.} \times 228.\dagger}{\text{R. P. M.}}$$

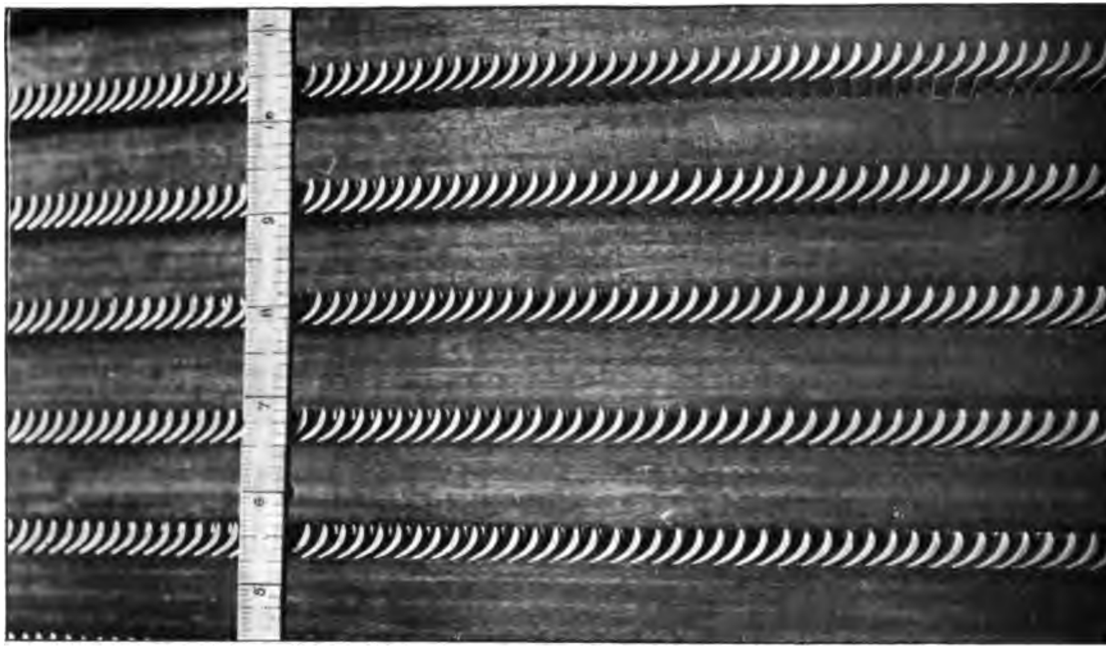
"To arrive at the corresponding number of rows, the revolutions being given, the ratio of V_t to V_s must be settled, from which the steam speed can be obtained; it is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis. The number of rows N on one diameter can be found by working out the B.Th.U.'s necessary to give a certain steam speed at each row, see Fig. 8, the available energy divided by the energy it is desired to abstract at each row will give the number of rows required. This result may be arrived at by various ways, but the principle involved is the same in each case. Numerous empirical coefficients for approximating steam speeds and the corresponding number of rows are obtainable from experience, and are similar in use and value to the Admiralty coefficient, that is, while they represent a crude method of doing something that should be done more scientifically, they are very simple and capable of rapid handling. Being, however, based on long and costly experiments, much reticence is observed regarding their publication. Varying, of course, with the steam pressure and vacuum, the number of rows on one diameter would involve an excessive length of turbine and also inconvenient blade heights. It is, therefore, usual to divide the rotor into three or more stages, which have the advantage of shortening the turbine and reducing the number of rows. If n = the fraction of power developed in

* Constant $223 = \sqrt{64 \times 778}$.

NOTE.—64 = gravity × 2; 778 foot-pounds = 1 B.T.U.

† Constant $228 = (60 \times 12) \div 3.1416$.

NOTE.—60 seconds = 1 minute; 12″ = 1 foot.



View of Blades looking down on Rotor.

NOTE.—The scale of inches shown gives a fair idea of the blade widths and the distance apart of the various rows.

[*To face page* 22.

the first cylinder or barrel, $\frac{N}{n}$ = number of rows in the first barrel, and with the alteration of diameter and increase of blade velocity in the succeeding stages, the number of rows on other barrels are so altered as to keep, for equal powers and efficiencies—

$$(\text{Blade velocity})^2 \times \text{No. of rows} = \text{Constant.*}$$

"The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and I think the Westinghouse Company at Pittsburg was first to make a radical departure in this and adopt far higher speeds. The maximum vane speed used for Parsons' blading is, as far as the author is aware, about 375 feet per second in the low pressure blades, and 170 in the H.P. blades of electrical turbines; the lowest speeds used are in marine work, and are only about one-third of these. To some extent blade speed is governed by blade height; the speed should be so modified that this may be at least 3 per cent. of the mean diameter to reduce the proportion of clearance losses. Leakage over the tips of the blades is perhaps not so detrimental on account of actual leakage loss as in its superheating effect on steam between the row past which it leaks and the last row, because this reheating effect upsets calculations regarding openings by increasing the steam volume, and thereby affects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is to resist distortion due to heat the less it may be made. . . .

"TABLE II.—MARINE WORK.

TYPE OF VESSEL.	Peripheral Vane Speed per Second. H.P.	L.P.	Mean Ratio of V_t—V_s	Number of Shafts.
High speed mail steamers	70- 80	110-130	.45-.5	4
Intermediate do.	80- 90	110-135	.47-.5	3 or 4
Channel steamers	90-105	120-150	.37-.47	3
Battleships and large cruisers	85-100	115-135	.48-.52	4
Small cruisers	105-120	130-160	.47-.5	3 or 4
Torpedo craft	110-130	160-210	.47-.51	3 or 4

"In Table II., the vane speeds adopted in various classes of work are given, and the reduction in peripheral speed on account of the propeller reducing the revolutions, and the necessary proportion of blade height modi-

* NOTE.—The "constant" referred to above appears to vary from 1400000 to about 1600000.

fying the diameter may be clearly seen. To this combined action is due the fact that only in the faster classes of vessels, or in those small types in which some propulsive efficiency can be sacrificed, is the turbine applicable. In slow cargo steamers, though the revolutions may be high enough, the power required is not sufficient to enable a reasonable blade height to be adopted, and it is this consideration—viz., proportion of leakage over blade tips—that curtails the wider adoption of this type of turbine. For the same low peripheral blade speed, other types of turbine are unsuitable on account of the impossibility of reducing the steam velocity sufficiently without abnormal weight and inefficiency.

"The smallest size of marine turbine is usually larger than the average electrical turbine as far as power is concerned, and therefore does not meet with the same commercial considerations as the smaller sizes of the latter type. These are not designed for the same internal efficiency as the larger machines,

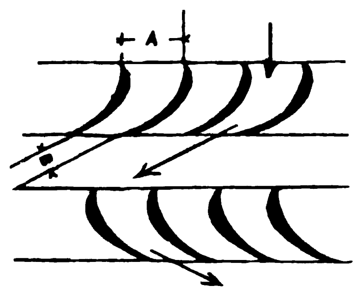


Diagram showing Area through normal Blades to allow passage of Steam, compared to Blade Annulus Area.

$B = A \div 3.$

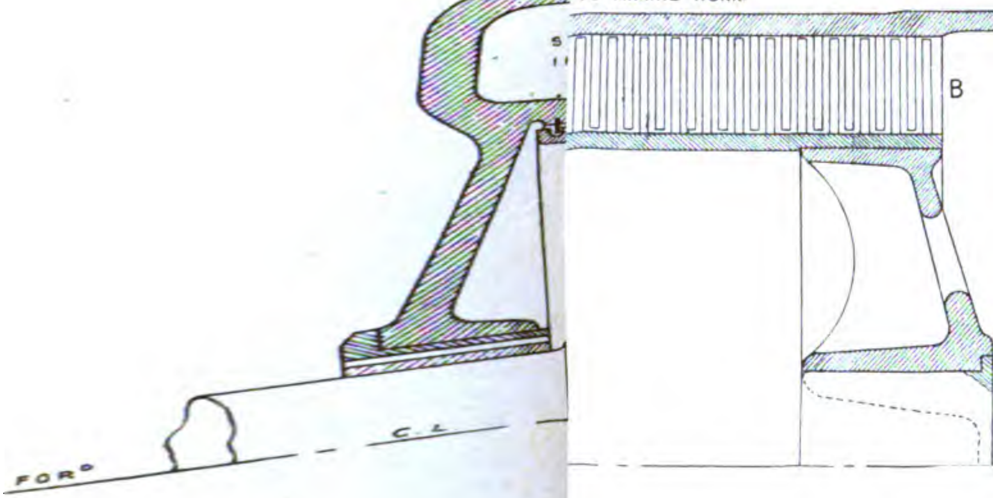
The area B depends on the steam velocity and steam volume, being more for low-pressure steam than for high-pressure steam.

chiefly on account of manufacturing cost, and they do not attain anything like the same efficiency compared with the Rankine cycle.

"Speaking in reply to the discussion on his paper to the Institution of Naval Architects in 1903, Mr Parsons said that, 'for all practical purposes, while the steam is traversing each set (of blades) as shown, it behaves like an incompressible fluid, just like water would do, as the expansion is very small at each set. The frictional losses and the eddy-making losses would be practically identical within small limits with what they would be with water, and the actual forces would be in proportion to the density of the medium. . . . In the turbine blades themselves, the efficiency is between 70 and 80 per cent.'

"Using this hydraulic analogy enables us to calculate the number of stages required in a different manner: the 'equivalent head,' due to the steam pressure, may be found, together with that at each row necessary to give the required velocity, from which both the number of stages and the coefficient of expansion at each stage may be worked out.

TO MARINE WORK.



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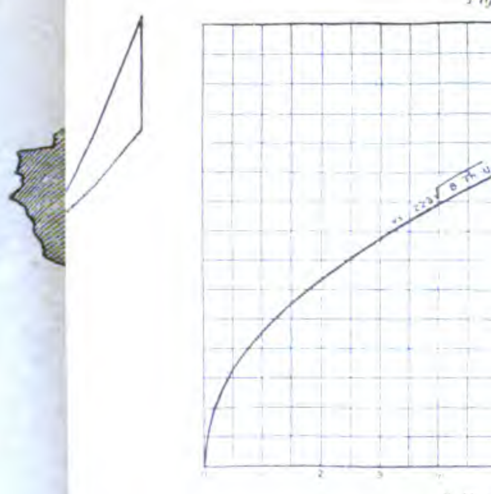


"In the early marine designs, such as the 'Queen Alexandra' and H.M.S. 'Amethyst,' the turbine drums were all made of the same diameter, and the higher speed necessary on the L.P.'s was got by running at considerably higher revolutions than on the H.P. shaft; but, following up the increase in propeller efficiency found to be due to the use of larger screws, the speed for each shaft is now more nearly equal, while the wing drums are made larger in diameter. The vagaries of the following wake, however, necessitate slightly different propeller dimensions on each shaft, or else slightly different revolutions with the same screws; and it is noticeable that in a triple-screw arrangement, the centre screw being right-handed and the wing screws revolving outwards, that the starboard propeller is influenced by the centre one, and almost invariably revolves at a lower speed. In a four-shaft design, due to the varying wake values at different speeds, and possibly, also, to some unequal distribution of power, the outer screws run slower at low speeds and faster at high speeds than the two inner shafts, but exact data as to this, and the possibility of allowing for it in the design, are still wanting.

"In all types of turbines—Parsons', Rateau's, Curtis', &c.—a certain ratio must be maintained between the blade velocity and steam velocity, and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by the revolutions or by large diameters, or both. As the weight increases very rapidly with the diameter, and extraordinarily so with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter, and especially should this be done in cases where, as in the Rateau or Zoelly types, the weight increases more rapidly in inverse proportion to the R.P.M. and the diameter than it does with other types. To increase the revolutions, it may be necessary to increase the number of shafts and propellers, thus reducing the power per shaft and the effective thrust through each screw. Increasing the diameter of the turbine adds largely to the constructional difficulties, especially of the cylinder.

"Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The height of blade depends on the volume of the steam and the speed at which it is to flow, and also on the ratio of the area of exit openings between the blades to that of the annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area to pass the steam being equal to volume ÷ velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for $\frac{1}{2}$, &c.), then

$$\text{Height of blade in inches} = \frac{\text{Clear area in square inches} \times 3}{\text{Mean circumference in inches}}.$$

"The ratio of blade height to mean diameter should not be less than 3 per cent. or more than 15 per cent., because in the former the leakage will be excessive, and in the latter the bending moment on the blade becomes too great, and the radial divergence of the blades too much. The width of blade, the shape of section adopted, and the circumferential pitch, are standard considerations, and affect the factor 3 given above. It is not proposed to enlarge upon them in this paper. It may, however, be remarked that for $\frac{V_t}{V_s}$ greater than .6 the usual shape of Parsons' section, as shown in Fig. 5, should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of

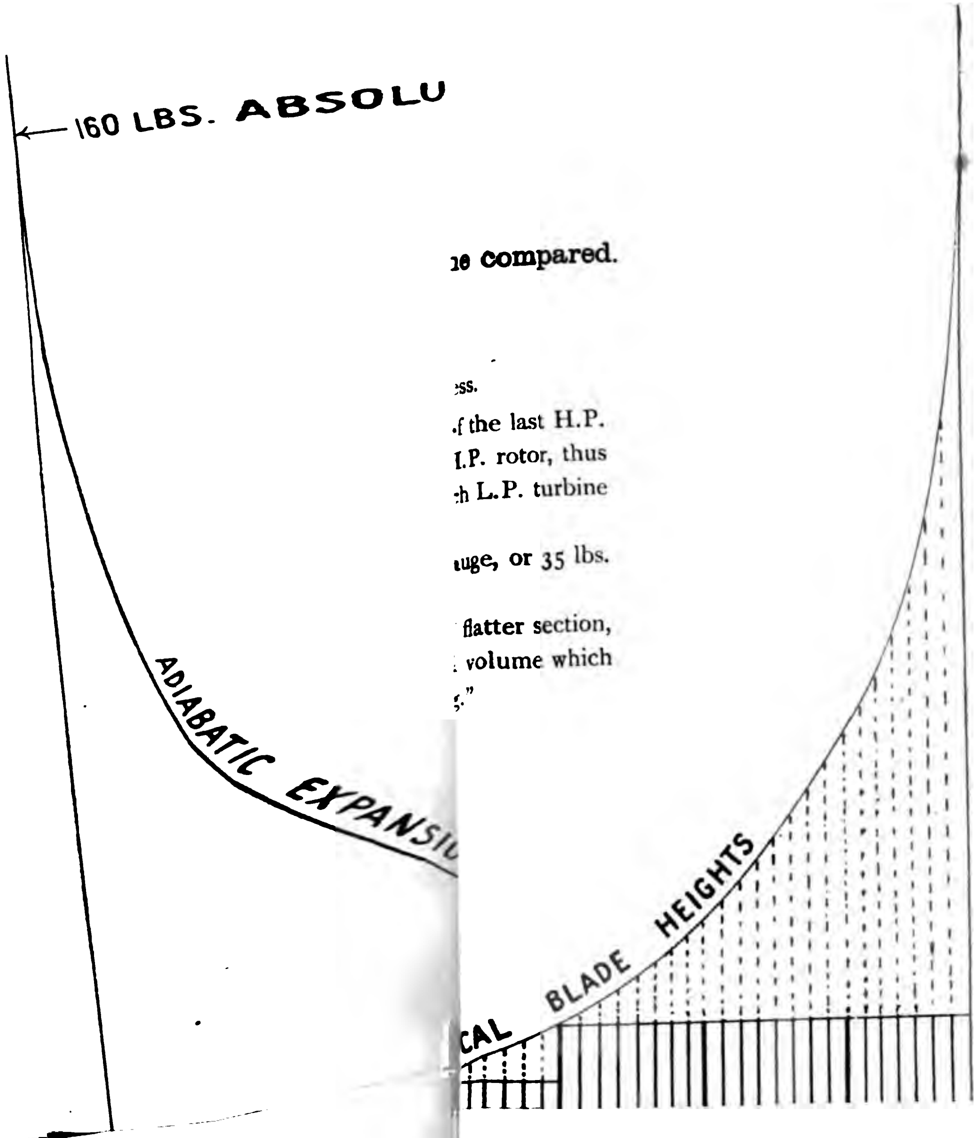
strengthening the blade sufficiently, the metal must be placed nearer the exit edge thus increasing the angle between the face and the back of the exit edge of the blades, and giving, in fact, an inferior shape of opening compared with that obtainable with a blade section adapted to ratios under .6. If, for the present, it is sufficient to use the blade sections and packing pieces similar to those now adopted so generally, in Table III. can be found a list of widths for a given height, and the axial spacing of the rows. While this must be kept down to reduce the length of drum, it must be sufficient to allow for some play in overhauling; and sufficient clearance can be allowed here without affecting the economy. The openings between the blades to allow of the passage of the steam are very important, and must be carefully designed. The actual volume of the steam—not the volume per lb., as found in tables, or the volume due to adiabatic expansion, but the exact volume per lb. at any point along the turbine—must be determined, in order to arrive at the desired adjustment of velocities. It is extremely doubtful whether the present blading arrangements give the best results; greater accuracy of calculation, and consequently improved pressure distribution and efficiency, seem likely to follow the use of a more mechanical blading construction."

Revolutions and Rotor Diameter.—The highest theoretical efficiency of the turbine is attained when the linear velocity of the rotating blades is about equal to one-half the velocity of the steam impinging upon those blades, and as this figure is very high there remain but two alternatives to obtain the results which the best efficiency would require. The first of these two is to arrange the revolution speed so high as to enable the vanes to receive the steam under the conditions stated above. The other alternative is to reduce the speed of rotation of the turbine by increasing its diameter in equal ratio to the reduction of rotative speed, which, of course, has the disadvantage of increasing the weight of turbines required.

Theoretical Blade Heights.—The next diagram shows how the blade heights would vary if made to exactly correspond to the steam expansion.

As the pressure falls the volume increases (adiabatically), so that each succeeding row of blades should be slightly longer than the preceding row. In practice, however, this is not carried out, the blades being arranged in sets of equal height, or, as it is called, "stepped." The horizontal dotted lines show the actual arrangement of blade heights as usually fitted. The 6th, 7th, and 8th expansions of each L.P. turbine consist of rows of blades of equal height, but it should be noted that the angle or curvature of each set is different, the 7th and 8th expansions having "wing" blades of flatter section and of wider circumferential pitch than those of the preceding sets.

It will be observed that the first expansion of the L.P. has *shorter* blades than the last expansion of the H.P. It must, however, be remembered that the diameter of the L.P. drums is *more* than the H.P., thus giving a higher peripheral and steam speed, also that in the usual standard arrangement of one H.P. and two L.P. turbines only *half* the quantity of steam passes through each L.P. turbine.



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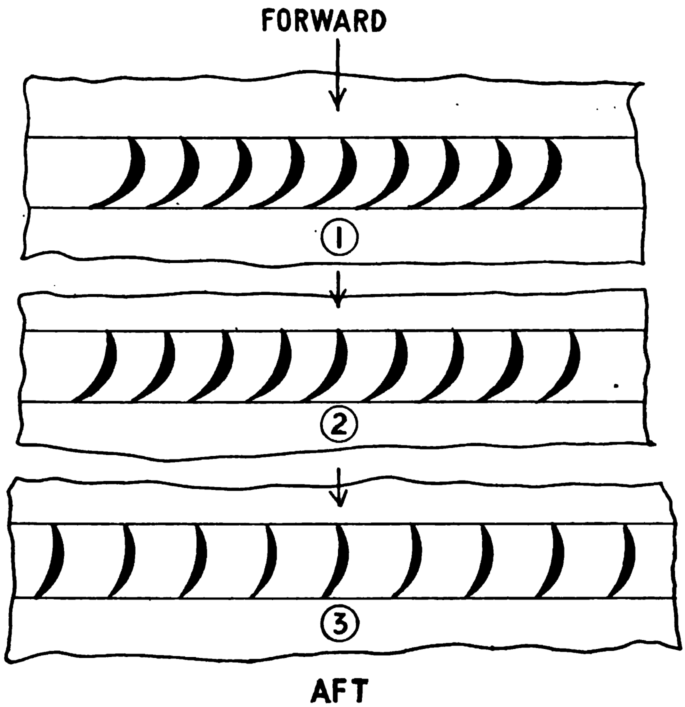
NOTE.—Observe that the total number of blades is generally the same for the H.P. turbine and each L.P. turbine, the H.P., however, only having four expansions, each one containing *twice* the number of blades contained in each of the eight expansions of the L.P. turbines.

Expansion Clearance.—At each change of expansion a clearance space is arranged between the last ring of blades of one set, and the first ring of the next set (see sketch) to allow of drop of pressure.



Variation in Blade Angles and Pitch in last three L.P. Expansions.

1. Sixth Expansion. 2. Seventh Expansion. 3. Eighth Expansion.

NOTE.—The above are examples of "wing" blades.

Blade Variation.—In the largest size of blades which are fitted on the after or low pressure end of each turbine, the curve and pitch is varied so as to really constitute three expansions, although the *height* of the blades is the same. The sketch above shows the variation of blade curvature and pitch, which is arranged to allow for the increasing volume of steam passing through the last section. If the last set of blades is made up of, say, 24 rings, all of the same height, then 8 rings are as of the 1st set, 8 of the 2nd set, and the remaining 8 as of the 3rd set. Observe that the blades are less curved aft, also that the pitch is increased, which results in a smaller

number of blades per ring. Theoretically each successive ring of blades throughout the whole turbine should be graded in this way.

NOTE.—In some cases the exit openings of the first half of the rows of each of the first few expansions have been reduced, which practically gives two expansions to what was formerly only one: this alteration increases the velocity of the steam passing between the rows with the restricted openings. A specially designed "closing-up" tool is employed to reduce the blade openings as described.

Velocity Calculations, &c.—The Foot-pounds of energy contained in a given weight of steam at a given pressure and velocity are found as follows:—

$$\text{Kinetic Energy} = \frac{W \times V^2}{64.4} \text{ Foot-pounds.}$$

NOTE.—W = Weight of steam in pounds.
V = Velocity of steam in feet per second.
64.4 = Acceleration due to gravity per sec. per sec.

The change in kinetic energy or the work done between the blades may be expressed as follows, friction and other losses, such as tip clearance leakage, being neglected.

$$\text{Kinetic Energy} = \frac{(V^2 - v^2) \times W}{64.4} \text{ Foot-pounds.}$$

Where V = Velocity of steam in feet per second at the entering edge of blades.
v = Velocity of steam in feet per second at the leaving edge of blades.
W = Weight of steam in pounds.
64.4 = Acceleration due to gravity per sec. per sec.

For example—Let V = 600 feet per second.
" W = 1 pound.

$$\text{Then, } \frac{W \times V^2}{64.4} = \frac{1 \times 600^2}{64.4} = 5590 \text{ Foot-pounds of kinetic energy.}$$

For example—Let V = 400 feet per second.
" v = 300 " "
" W = 1 pound.

$$\text{Then, } \frac{(V^2 - v^2) \times W}{64.4} = \frac{(400^2 - 300^2) \times 1}{64.4} = \begin{cases} 1087 \text{ Foot-pounds in passing} \\ \text{across one row of blades.} \end{cases}$$

NOTE.—In turbine practice the steam expands approximately adiabatically. No heat being supplied from without, and if no leakage of heat takes place, the work is done at the expense of the internal heat energy of the steam, and the fall in pressure, in temperature, and amount of condensation in the turbine is proportional to the work done.

The expansion being adiabatic, the heat drop for a given pressure drop is much more than that shown in the total heat table of saturated steam, as the steam which condenses during the performance

of work reduces the weight of actual steam remaining after expansion, and therefore the heat contained per pound of steam and water mixture is proportionally less. It should be again noted that the internal heat energy of the steam is transformed into mechanical energy, hence the heat drop so often referred to.

Let V = Velocity of steam.
„ W = Weight.
„ H = Heat units given up.
64.4 = 32.2 × 2 (acceleration due to gravity per sec. per sec.).

$$\text{Then, } \frac{V^2 \times W}{64.4} = \text{Foot-pounds of kinetic energy.}$$

$$\text{And, Foot-pounds} \times 64.4 = V^2 \times W.$$

$$\text{Therefore, } H \times 778 \times 64.4 = V^2 \times W.$$

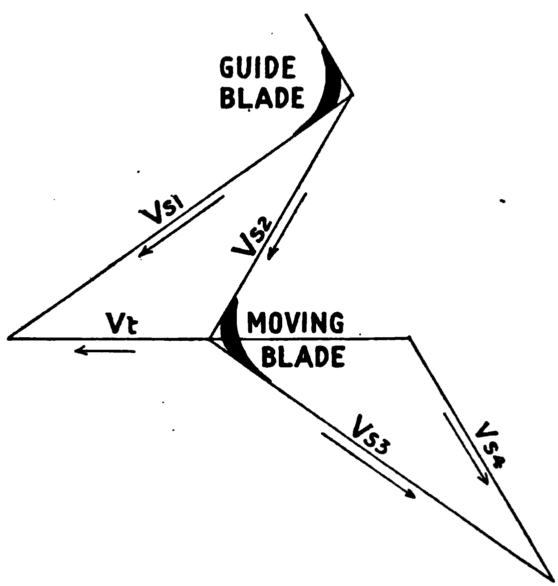
$$,, \quad V^2 = \frac{H \times 778 \times 64.4}{W}$$

$$,, \quad V = \sqrt{\frac{H \times 778 \times 64.4}{W}}$$

$$,, \quad \frac{V^2 \times W}{64.4} = H \times 778.$$

$$\text{And, } \frac{\text{Foot-pounds}}{778} = H.$$

Velocity Diagram.



The above diagram shows graphically the varying steam velocities previously referred to.

V_{s1} = absolute steam velocity (initial).
V_{s2} = relative " " " with regard to blade or rotor speed.
V_{s3} = relative " " leaving the blades.
V_{s4} = absolute " " " "
V_t = blade velocity.

The actual velocity of the steam passing between the rows of blades depends upon the drop of pressure produced by the expenditure of Foot-pounds of heat energy developed as work done.

The velocity of the steam in feet per second, due to any pressure drop, is found as follows:—

$$\text{Velocity of steam} = \sqrt{64.4 \times 778 \times (H_1 - H_2)}.$$

NOTE.—H_1 = Initial heat units.
H_2 = Final " "
64.4 = twice 32.2 (gravity acceleration per sec. per sec.).

EXAMPLE.—Calculate the velocity of the steam between the blades of a Parsons turbine if the heat units per pound of steam at admission edge are 1,200, and at exhaust edge 1198 units (a heat drop of 2 units, neglecting frictional and other losses).

$$\text{Then, } \sqrt{64.4 \times 778 \times (H_1 - H_2)} = \sqrt{64.4 \times 778 \times (1200 - 1198)}$$
$$= 316 \text{ ft. Velocity per second.}$$

EXAMPLE.—Calculate the energy contained in one pound of the above steam.

RULE.—$\frac{V^2 \times W}{64.4}$ = Foot-pounds; Therefore, $\frac{316^2 \times 1}{64.4}$ = 1556 Foot-pounds, and $1556 \div 778 = 2$ B.T. Units as before.

Absolute Velocity and Relative Velocity.—By absolute velocity is meant the velocity of the steam with regard to stationary objects.

By relative velocity is meant the velocity of the steam with regard to moving objects, in this case with regard to the blade velocity.

The two steam velocities referred to are shown graphically in the sketch, and it will be observed that the rotor velocity affects the relative velocity of the steam inversely, that is, with high rotor speed the relative steam speed on leaving the moving vanes is low, and *vice versa.*

Guide Blades.—In the guide blades, the work done in producing increase of steam velocity is equal to the following:—

$$\frac{V_{s1}^2 - V_{s4}^2}{64.4} = \text{Foot-pounds in guide blades.}$$

Moving Blades.—In any of the moving blades, the work done in producing shaft rotation is equal to the following:—

$$\frac{V_{s3}^2 - V_{s2}^2}{64.4} = \text{Foot-pounds in moving blades.}$$

Notice that the work done in the moving blades is calculated from the *relative* steam velocity of admission and *relative* steam velocity of exit, whereas the work done in the guide blades is calculated from the absolute steam velocities of both admission and of exit.

Work of Steam Acceleration.—The work done by the steam upon itself in the way of increased velocity in the first row of stationary or guide blades of a turbine is, as explained above, found as follows:—

$$\frac{V^2 \times W}{64.4} = \text{Foot-pounds.}$$

EXAMPLE.—Calculate the Foot-pounds of energy developed by one pound of steam within the guide blades of a turbine if the steam velocity is 300 feet per second.

$$\text{Then, } \frac{V^2 \times W}{64.4} = \frac{300^2 \times 1}{64.4} = 1400 \text{ Foot-pounds (nearly).}$$

In any turbine, then, the total energy per pound of steam supplied is equal in Foot-pounds to the following:—

$$(1114 + .3 \times T^\circ) \times 778 = \text{Foot-pounds contained.}$$

Therefore, after the steam has passed through a pair of rows of blades, the equation will be as follows:—

$$(1114 + .3 \times T^\circ) \times 778 = \frac{V^2}{64.4} + H \times 778.$$

Where H = heat left in the steam after fall in pressure between blade rows.

From the above it will be obvious that the total energy remains the same, and is exactly equal to the work done added to the energy still remaining in the steam, part of which will be represented by the small per cent. of water condensed, and the remainder by the actual steam velocity.

EXAMPLE.—At the inlet edge of the blades the pressure is 140 lbs., and at the outlet edge 138 lbs., the difference of B.T. units due to the nature of the expansion being $2.5 = (H_1 - H_2)$. Calculate (1) the steam velocity, (2) the kinetic energy of the steam.

$$V = \sqrt{64.4 \times (H_1 - H_2) \times 778}.$$

$$\text{Therefore, } \sqrt{64.4 \times 2.5 \times 778} = 354 \text{ feet per sec.}$$

$$\text{and, } \frac{V^2}{64.4} = \text{kinetic energy} = \frac{354^2}{64.4} = 1945 \text{ Foot-pounds.}$$

EXAMPLE.—The two velocities of the steam are $V_1 = 300$ feet per second, and $V_2 = 450$ feet per second. Calculate the Foot-pounds of work given up, and the number of B.T. units of heat drop across the blades per pound of steam.

$$\text{Kinetic energy} = \frac{(V_2^2 - V_1^2) \times W}{64.4} = \frac{(450^2 - 300^2) \times 1}{64.4} = 1746.8 \text{ ft.-lbs.}$$

$$\text{Heat drop} = \left(\frac{V^2 \times W}{64.4}\right) \div 778 = \left(\frac{450^2 \times 1}{64.4}\right) \div 778 = 4.04 \text{ units of heat.}$$

Notice that the relative velocity of the entering steam is less than its absolute velocity, and that the relative velocity of the steam leaving the moving blades is *more* than its absolute velocity. This is due to the velocity of the blades being in one constant direction.

Blade Velocity.—The mean velocity of the blades being often from .4 to .5 of the steam velocity, then the steam velocity × .5 = blade velocity at mean diameter of the blade circle. So that if the steam velocity is, as stated, 18000 feet per minute, then,

$$18000 \times .5 = 9000 \text{ feet per minute.}$$

Now if the revolutions are to be not more than, say, 400 per minute, the diameter of rotor can be found as follows :—

$$\frac{9000}{400 \times 3.1416} = 7.08 \text{ feet diameter across mean diameter of blades.}$$

The actual diameter of rotor would probably be somewhere under 7 feet, when the length of blades is deducted.

Rotor Diameter.—The difference in turbine diameter required for low and high revolution speed is shown in the following cases :—

Let V_s = steam velocity, and V_t = blade velocity.

EXAMPLE.—Steam velocity $V_s = 21000$ feet per minute, and if $\frac{V_t}{V_s} = .5$, find the diameter of rotor required for (1) 600 revolutions, and (2) 200 revolutions per minute.

Then, (1) $21000 \times .5 = D \times 3.1416 \times 600.$

$$\therefore \frac{21000 \times .5}{3.1416 \times 600} = 5.5 \text{ feet diameter of rotor.}$$

Again, (2) $21000 \times .5 = D \times 3.1416 \times 200.$

$$\therefore \frac{21000 \times .5}{3.1416 \times 200} = 16.7 \text{ feet diameter of rotor.}$$

NOTE.—The above diameters are measured across the mean *height* of blades, so that the actual diameter of rotor might be, say, 9 or 10 inches less than the above results.

Increase of Velocities.—At each succeeding expansion, the mean diameter of the blades being increased owing to increase of blade height, the velocity of the blades per second at the after or exhaust end of any turbine rotor is rather more than at the admission or forward end. An example will make this clear.

EXAMPLE.—Calculate the respective blade velocities at each expansion of the H.P. turbine, if the blade heights are as follows:—

$1\frac{1}{16}$ in. at 1st expansion.
$1\frac{3}{8}$ in. „ 2nd „
$1\frac{7}{8}$ in. „ 3rd „
$2\frac{1}{2}$ in. „ 4th „

The diameter of rotor is 3 feet 6 inches, and the revolutions per minute 600. Diameter of rotor = 3 feet 6 inches = 42 inches, then

At 1st expansion, 42 in. + $1\frac{1}{16}$ in. = $43\frac{1}{16}$ in. = 43.0625 mean diameter across blades.

At 2nd expansion, 42 in. + $1\frac{3}{8}$ in. = $43\frac{3}{8}$ in. = 43.375 mean diameter across blades.

At 3rd expansion, 42 in. + $1\frac{7}{8}$ in. = $43\frac{7}{8}$ in. = 43.875 mean diameter across blades.

At 4th expansion, 42 in. + $2\frac{1}{2}$ in. = $44\frac{1}{2}$ in. = 44.5 mean diameter across blades.

Therefore, $\frac{43.0625 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{112 ft. per second velocity of blades}\\ \text{at 1st expansion.}\end{array}\right.$

„ $\frac{43.375 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{113 ft. per second velocity of blades}\\ \text{at 2nd expansion.}\end{array}\right.$

„ $\frac{43.875 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{114 ft. per second velocity of blades}\\ \text{at 3rd expansion.}\end{array}\right.$

„ $\frac{44.5 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{116 ft. per second velocity of blades}\\ \text{at 4th expansion.}\end{array}\right.$

NOTE.—Divide by 60 seconds and by 12 inches to obtain feet per second.

NOTE.—The blade tip clearance, about $\frac{50}{1000}$ of an inch, or .05 in., is neglected in the above calculations.

It should be noted that the steam velocities increase correspondingly at each successive expansion, so that the average ratio of blade velocity to steam velocity, or $\frac{V_t}{V_s}$, is maintained more or less constant throughout the turbine.

From the foregoing, it will be easily seen that the blade heights and openings must increase at a high ratio to allow for the very rapid increase of volume at the lower pressures carried near the exhaust end of the low-pressure turbines.

Steam Velocity.—The velocity or speed of steam at any given pressure varies according to the pressure of the exhaust or opposing back pressure. In marine turbine practice, the velocity of the steam is often about 300 feet per second, or 18,000 feet per minute.

For example, at an initial pressure of 180 lbs. the velocity of steam when flowing directly into the atmosphere is equal to about 3,000 feet per second, and if allowed to flow into a vacuum of 25 inches the velocity is about 3,700 feet per second.

After passing each row of rotor blades a small drop in pressure and in heat energy takes place owing to the useful work done by the steam in rotating the shaft and in increasing its own velocity. This drop is proportional to the foot-pounds of energy expended at each row of blades, and, as before stated, the exchange of heat into work or kinetic energy is equal to 778 foot-pounds per unit of heat. The average drop in pressure per row of blades throughout a marine turbine works out as about .76 of a pound.

Heat Drops.—The fall of pressure, temperature, and heat units in the casing of a turbine may be described in a somewhat rudimentary form as follows:—As previously explained, the necessary steam velocity required to produce effective work, or kinetic energy, on the blades of the turbine rotor is obtained by allowing a suitable drop in pressure and in heat units between the successive pairs of rows of blades. It is important to note that the average drop per pair of rotor blade rows is less in the L.P. turbines than in the H.P. turbine. This is accounted for by the fact that the difference in total heat units of steam at low pressure is more than that of high-pressure steam for the same pressure drop, or in other words, an equal difference in drop of heat units can be obtained by a smaller drop of pressure. The difference in drop of heat units necessary for the required foot-pounds of energy to be developed can therefore be obtained by a smaller pressure drop per pair of rows with low-pressure steam, as the following examples will perhaps make clear:—

Work done during Adiabatic Expansion.—To calculate the work done, or, which is the same thing, the units of heat given up or converted into work during the adiabatic expansion of steam in a turbine, the following data are required:—

The absolute temperature of the steam before and after expansion.
The latent heat of the steam before and after expansion.
The dryness factor of the steam before and after expansion.

Let T_1° = Absolute temperature before expansion.
T_2° = " " after "
H_1 = Latent heat before expansion.
H_2 = " " after "
f_1 = Dryness factor before expansion.
f_2 = " " after "

The heat energy given out in British Thermal Units

$$= f_1 \times H_1 - f_2 \times H_2 + T_1^\circ - T_2^\circ = \text{B.T.U.}$$

And, $\text{B.T.U.} = \dfrac{V^2}{2g \times 778}$; or, $V^2 = 2g \times 778 \times \text{B.T.U.}$

Therefore, $V = \sqrt{2g \times 778 \times \text{B.T.U.}}$

NOTE.—V = Velocity of steam in feet per second.
$2g = 2 \times 32.2$ = Acceleration due to gravity in feet per second per second.

NOTE.—The "Entropy" diagram affords the clearest explanation of the heat utilisation and expenditure.

EXAMPLE 1—**High-Pressure Steam.**—In a marine turbine find the heat energy given up by one pound of steam when flowing from one expansion where the pressure is 150 lbs. absolute, to another expansion where the pressure is 140 lbs. absolute, given that—

$$\begin{aligned}T_1^\circ &= (358^\circ + 461^\circ) = 819.\\ T_2^\circ &= (353^\circ + 461^\circ) = 814.\\ H_1 &= 861.\\ H_2 &= 865.\\ f_1 &= 1.\\ f_2 &= .996.\end{aligned}$$

NOTE.—Absolute temperature = Fahrenheit + 461°.

$$\text{Heat units given up} = 1 \times 861 - .996 \times 865 + 819 - 814$$
$$= 861 - 861.54 + 819 - 814 = 1680 - 1675.5 = 4.5 \text{ B.T.U.}$$

Also, calculate the velocity of the steam.

$$\text{Then, } V = \sqrt{64.4 \times 778 \times 4.5} = 474 \text{ feet per second.}$$

EXAMPLE 2—**Low-Pressure Steam.**—In a marine turbine find the heat energy given up by one pound of steam when flowing from one expansion where the pressure is 6 lbs. absolute, to another expansion where the pressure is 2 lbs. absolute, given that—

$$\begin{aligned}T_1^\circ &= (170^\circ + 461^\circ) = 631^\circ.\\ T_2^\circ &= (126^\circ + 461^\circ) = 587^\circ.\\ H_1 &= 995.\\ H_2 &= 1026.\\ f_1 &= .85.\\ f_2 &= .8.\end{aligned}$$

$$\text{Heat units given up} = .85 \times 995 - .8 \times 1026 + 631 - 587$$
$$= 845.75 - 822.8 + 631 - 587 = 1476.75 - 1409.8 = 66.95 \text{ B.T.U.}$$

From the foregoing it will be seen that in Example No. 1, with a pressure drop of 10 lbs. (from 150 to 140 lbs.) and the conditions stated, the number of heat units converted into work, or kinetic energy, is only 4.5, whereas in Example No. 2, with a *pressure drop of only* 4 *lbs.* (from 6 lbs. to 2 lbs. absolutely), the number of heat units converted into kinetic energy is 66.95, thus clearly indicating the greater value of lower pressure steam in turbine practice, and accounting for the two L.P. wing turbines each developing equal power to the centre H.P. turbine with about half the weight of steam at a very much lower pressure.

The B.T.U. or heat drop for a given pressure drop *increases with the fall of pressure*, and as the kinetic energy given up to the blades entirely depends on the *heat drop*, it naturally follows that in the case of very low-pressure steam the same amount of work can be obtained with a much smaller pressure drop, as also shown in the example from actual practice given on page 142. It should be carefully noted that in turbine practice the kinetic energy got out of

the steam at each stage or pair of rows varies directly as the drop in heat units, and is, strange as it may appear, quite independent of the pressure carried. It will also be obvious that a high vacuum will increase the efficiency of the low-pressure turbines by allowing of a further drop in pressure, thus developing to the full extent the benefit of the increasing drop in heat units per given pressure drop. Allowing, therefore, equal powers to be developed in each shaft and in each turbine, it is of interest to note that—

1. The drop of pressure is less throughout each L.P. turbine to develop the same power as in the H.P. turbine.

2. The amount of steam used in each L.P. turbine is roughly only half that used in the H.P. turbine to develop the same power, as the exhaust from the H.P. turbines divides into two portions, one for each L.P. turbine.

Work done by Difference of Vacuum in L.P. Turbines.—When running at reduced speed with all turbines working (as will be noticed in the example given on page 170), the H.P. turbine carried an initial pressure of 80 lbs. by gauge, and the L.P. turbines each an initial pressure corresponding to 15 in. vacuum, which, worked out, gives an absolute pressure of $7\frac{1}{2}$ lbs., as 15 in. $\div 2 = 7.5$ lbs. vacuum, and 15 lbs. (atmospheric pressure), less 7.5 lbs., is equal to 7.5 lbs. absolute initial pressure. From this it will be apparent that each L.P. turbine, working between an initial vacuum of 15 in. and exhaust vacuum 28 in., develops the *same power* as the H.P. turbine, with an initial pressure of 80 gauge and an exhaust pressure of 7.5 lbs. absolute.

So that,

H.P. turbine	initial pressure	$= 80 + 15 =$	95 lbs.	absolute	pressure.
„ „	exhaust „	$=$	(say) $8\frac{1}{2}$ lbs.	„	„
L.P. turbines	initial „	$=$	7.5 lbs.	„	„
„ „	exhaust „	$=$	(say) 2 lbs.	„	„

NOTE.—The actual back pressure against the H.P. turbines will be about one pound or so in excess of the L.P. initial pressure, and the back pressure against the L.P. turbines about a pound in excess of the condenser pressures.

Referring to the above figures, it will be evident that each L.P. turbine, working with a total pressure drop of $7.5 - 2 = 5.5$ lbs. only, develops or gives out fully the same total heat drop, and therefore the same developed power as the H.P. turbine, working at a total pressure drop of $95 - 8.5 = 86.5$ lbs. This fact brings out, in a very striking manner, the high heat value of very low-pressure steam when applied to steam turbine practice, and affords perhaps the greatest contrast of all to reciprocating engine practice.

Pressure Drop in H.P. and L.P. Turbines.—The pressure drop per row is less in the L.P. turbines than in the H.P., as will be seen by referring to the practical example given on page 142, and the great difference in volume of one pound of steam at high pressure

and one pound of steam at low pressure is shown by the following figures :—

H.P. Turbine.

At 158 lbs. pressure absolute the volume is 2.81 cub. ft.
„ 160 lbs. „ „ „ 2.78 cub. ft.

The difference = .03 cub. ft. { for a pressure drop of 2 lbs.

L.P. Turbines.

At 10 lbs. pressure absolute the volume is 37.84 cub. ft.
„ 10.5 lbs. „ „ „ 36.14 cub. ft.

The difference = 1.70 cub. ft. { for a pressure drop of .5 lb.

The above demonstrates the necessity for longer blades and wider spacings at the exhaust ends of the L.P. turbines.

Drop of Pressure.—Between each pair of rows of blades the steam drops a certain amount in pressure, and the approximate drop can be shown as follows :—Suppose the initial pressure in the H.P. turbine to be 140 lbs. gauge, and the terminal pressure at the last row of blades to be, say, 22 lbs. gauge, then,

$$140 - 22 = 118 \text{ lbs. total drop of pressure in H.P. turbine.}$$

If, then, the H.P. turbine is made up of, say, 60 rows of blades in all, and we divide the total drop by the total number of rows, we obtain the average drop at each row, thus—

$$118 \text{ lbs.} \div 60 \text{ rows} = 1.96 \text{ lb. average drop per row.}$$

NOTE.—It should be noted that there will also be 60 rows of fixed or casing blades.

Again, take the L.P. turbine, also formed of 60 rows of blades, the initial pressure being, say, 20 lbs. gauge, and the condenser back pressure $1\frac{1}{2}$ lbs. (27 in. vacuum), or, say, $2\frac{1}{2}$ lbs. absolute actual pressure at last row of L.P. blades.

Then, $20 + 15 = 35$ lbs. absolute initial pressure,
and $35 - 2.5 = 32.5$ total drop in pressure in L.P. turbine:
then, $32.5 \div 60 = .54$ of a lb. average drop per row.

It should be noted that this drop takes place simultaneously in each L.P. turbine, as the H.P. exhaust divides into two branches, one for each L.P. turbine. The pressure drop gradually *decreases* from the initial end to the exhaust end of each turbine.

Increasing Steam Velocity.—The velocity acquired by the steam at any given position of the rotor entirely depends on the drop produced by the work done or loss of energy at each row, and as the steam expands in falling in pressure it follows that the area allowed for steam flow must be increased to give equal velocities throughout the turbine. As, however, a Parsons type turbine is composed of a

certain number of rows of blades, all of the *same size*, for each expansion, the velocity of the steam at the after rows will be more than that at the forward rows.

To take a practical example. A certain Parsons marine turbine consists, say, of the following:—

H.P. turbine	= 4 expansions, each containing	16 rows	= 64 rows.
L.P. ,,	= 8 ,, ,, ,,	8 ,,	= 64 ,,
L.P. ,,	= 8 ,, ,, ,,	8 ,,	= 64 ,,
		Total number of rows of blades,	192

Therefore, if the steam velocity at the first row of the H.P. turbine is, say, 250 feet per second, at the last or sixteenth row of that expansion the velocity will be more, as the volume, due to pressure drop, is increased, with the same blade heights and openings. This holds good for each of the various expansion sets or rows, and the increase of steam velocity can be approximately determined by comparing the volume of steam per pound for each of the two pressures, initial at first row, and terminal at last row.

The difference in blade heights is more marked in the L.P. turbines, as the volume increases very rapidly with fall of pressure in the case of low-pressure steam. The following figures will perhaps make this clear:—

Steam Pressures and Volumes.

Pressure (Absolute).	Cubic feet of Steam per pound weight (Steam volume).	Pressure (Absolute).	Cubic feet of Steam per pound weight (Steam volume).
210 lbs.	2.15 cubic feet.	55 lbs.	7.61 cubic feet.
195 ,,	2.31 ,,	35 ,,	11.64 ,,
175 ,,	2.55 ,,	15 ,,	25.84 ,,
155 ,,	2.87 ,,	10 ,,	37.84 ,,
135 ,,	3.26 ,,	8 ,,	46.68 ,,
115 ,,	3.80 ,,	6 ,,	61.20 ,,
95 ,,	4.54 ,,	4 ,,	89.63 ,,
75 ,,	5.68 ,,	2 ,,	172.08 ,,

Notice that one pound of steam at 210 lbs. absolute pressure occupies a volume of 2.15 cubic feet, and one pound at 2 lbs. absolute pressure a volume of 172.08 cubic feet.

Pressure Drop, Number of Rows, and Revolutions.—For a given diameter of rotor and required turbine efficiency—

1. A small pressure drop per row produces a low steam velocity, demanding a corresponding low revolution speed, and requiring a large number of rows to absorb the available heat energy of the steam.

2. A large pressure drop per row produces a high steam velocity, demanding a corresponding high revolution speed, and requiring a smaller number of rows to absorb the available heat energy of the steam.

NOTE.—In No. 1 case the revolution speed may be reduced if the diameter of the rotor be increased (as described on page 32), as this will also give the required high peripheral blade speed necessary to still maintain the ratio of $\frac{V_t}{V_s}$ and give constant turbine efficiency.

From 8 lbs. pressure (absolute) down to 2 lbs. pressure (absolute) the increase of steam volume per pound is very marked, and complicates the design of the L.P. turbines, owing to the necessity for allowing suitable blade heights and openings to prevent rapid increase of steam velocity at the low pressures referred to. The diagram of Blade Heights and Steam Volumes compared, facing page 26, illustrates this point to some degree.

Number of Blades per Row.—Regarding the number of blades per row it is of importance to note that in the H.P. turbine and reverse turbines the total number of blades in each row of the rotor usually exceeds the total number in each corresponding row of the casing, and the arrangement is carried out throughout the turbine or cylinder, whereas, in the L.P. turbines, the number of rotor blades in each row of the first four expansions only generally exceeds the number of blades in the casing rows, as throughout the remaining four expansions the number of rotor blades per row is usually less than those in the casing, this being necessary to allow of the exceptionally large volume of steam produced at the last L.P. expansions to flow through the moving blades without attaining an excessive velocity.

Special tools are now being used either to "close up" or "open out" the exit edges after the blades are already fixed in position, as it is upon the correct adjustment of the blade exit area that the efficiency and economy of the turbines greatly depend, and evidently the only true guide to which is actual experiment, theoretical calculations evidently being in many cases unreliable and misleading, as it is quite a common experience for the blade openings to be increased after the turbines are bladed.

Tip Leakage and Dryness Factor.—Owing to blade tip leakage, the adiabatic expansion of the steam may be said to be incomplete, as the leaking steam partly superheats the expanded and lower pressure steam further on, thus raising the dryness factor above the calculated amount obtained by pure adiabatic expansion alone, and proportionally reducing the heat drop and work done. A reference to the "error" diagram and "Table of Actual Volumes," page 41, together with the Rule for work done by adiabatic expansion on page 34, will perhaps make this clear.

Turbine Efficiency.—The principal turbine losses are usually estimated as follows :—

Friction of steam against blades,
" " " itself, } 20 per cent.

Leakage over blade tips, 8 per cent.

" " dummy piston at initial end of turbines,
" " steam glands at both ends of turbines,
Exhaust steam heat loss, } 5 per cent.

Mechanical friction of bearings, &c., 8 per cent.

These losses in efficiency have been found to total up to somewhere about 40 per cent., thus leaving an actual mean efficiency of 60 to 62 per cent.

To express the efficiency another way.

Steam consumption in theory ÷ steam consumption in practice =.60 to .62.

From various cases taken from actual practice the above figures work out as the average efficiency. The steam and blade friction losses are lowest at the H.P. turbine, and highest at the L.P. turbines, owing to the increase of friction at the longest blade heights.

Actual Volume of Steam.—According to the "error" diagram of steam volumes supplied by Mr E. M. Speakman (see page 21), the actual steam volume or dryness value in the turbine is less than the specific volume by from about 4 per cent. at 80 lbs. absolute pressure to 20 per cent. at 2 lbs. absolute pressure, the volume of steam in the turbine occupying to a certain degree an intermediate position between the specific volume and the volume due to adiabatic expansion. The actual steam volume can therefore be determined as follows :—

$$\text{Actual Steam Volume} = \frac{\text{Specific Volume} \times 100}{100 + \text{per cent difference}}.$$

EXAMPLE.—Determine the actual steam volume per pound in an L.P. turbine if the pressure is 15 lbs. absolute, and the error difference 11 per cent. (by diagram, page 21).

Then, 15 lbs. absolute = 25.84 cub. ft. Specific Volume (from Steam Table),

$$\text{and, } \frac{25.84 \times 100}{100 + 11} = 23.27 \text{ cubic feet Actual Volume.}$$

NOTE.—It will be seen by referring to the error diagram that the actual steam volume at the above pressure is also fully 4 per cent. *more* than that due to adiabatic expansion.

Actual Steam Volumes in Turbine.—The following table of actual steam volumes per pound and dryness factors to correspond is calculated from the error diagram given by Mr E. M. Speakman in his paper on "The Steam Turbine with Special Reference to Marine Work," and gives a fair idea as to the required relative proportions between blade heights, blade angles, and steam speeds at

the various pressures existing in the turbine casings at the different expansions.

Table of Steam Volumes.

Absolute Pressure in lbs.	Specific Volume in Cubic Feet per lb.	Actual Volume in Turbine in Cubic Feet per lb.	Dryness Factor.
100	4.33	4.18	.965
80	5.34	5.12	.958
60	7.01	6.66	.950
55	7.61	7.19	.945
50	8.32	7.84	.943
45	9.19	8.64	.940
40	10.26	9.61	.936
35	11.64	10.85	.932
30	13.46	12.46	.925
25	15.97	14.65	.917
20	19.71	17.91	.908
15	25.84	23.23	.898
10	37.84	33.48	.884
9	41.77	36.80	.881
8	46.68	40.94	.877
7	52.93	46.22	.873
6	61.20	53.10	.867
5	72.99	62.92	.862
4	89.63	76.60	.854
3	117.50	99.66	.848
2	172.08	143.70	.835
1.5	225.58	185.66	.823

Referring to the error diagram mentioned (page 21), it will be seen that at, say, 10 lbs. pressure absolute, the actual steam volume in the turbine is less than the specific volume of dry steam by about 12½ per cent., and more than the volume due to adiabatic expansion by fully 5 per cent.

Velocity Triangles and Horse-Power.—The work done per pound of steam flow and the horse-power developed depends on the initial velocity and exit velocity of the steam, together with the efficiency of the blades.

The steam exit velocity and mean blade velocity being known, the initial velocity of the steam can be determined by the geometrical construction of a velocity triangle, given, of course, the blade angle of the "expansion" under consideration.

The exit angle given to the blades largely determines the exit openings available for steam flow, and therefore governs the exit

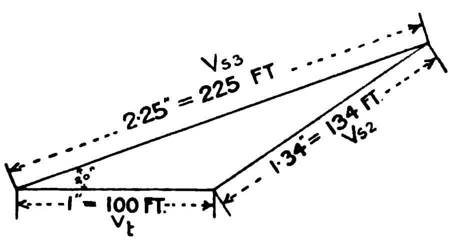
steam velocity; this, again, together with the blade velocity, regulates the admission velocity of the steam, and consequently the work done per blade row. The actual areas of exit openings can be accurately determined by mathematical calculations, or by actual measurement, after which the steam exit velocity can be found if the weight of steam flow and actual volumes of the mixture (steam and water) are known. From these, a "velocity triangle" can be constructed, and the theoretical velocity of steam admission determined.

EXAMPLE 1.—Construct a velocity triangle, and determine the horse-power developed at the first H.P. expansion of 16 rows, given the following:—

Mean diameter across blades - - -	48 inches.
Revolutions - - - - - - -	480 per minute.
Steam exit velocity - - - - -	225 feet per second.
Steam flow per minute - - - -	2,400 lbs.
Assume blade efficiency as - - -	85 per cent.
Scale of triangle - - - - -	100 feet per inch.
Blade angle - - - - - - -	20° (normal blades).

$$\text{Then, } V_t \text{ (blade velocity)} = \frac{48 \times 3.1416 \times 480}{12 \times 60} = 100 \text{ feet per second.}$$

Set off the angle of 20° as shown, and measure up by the scale the steam exit velocity of 225 feet, which will be equal to 2.25 inches, then set off horizontally the blade velocity of 100 feet, equal to 1 inch; now complete the triangle, and the third side, which will be found to measure 134 feet (or 1.34 inches), represents the initial velocity of the steam.



Velocity Triangle H.P. Turbine.

The work done in Foot-pounds per minute in this expansion =

$$\frac{(225^2 - 134^2) \times 2400 \times 16 \text{ rows} \times .85}{64.4} = 16557704 \text{ foot-pounds.}$$

$$\text{Horse-power} = 16557704 \div 33000 = 501.7.$$

EXAMPLE 2.—Repeat the foregoing for the first L.P. expansion of 8 rows, given the following:—

Mean diameter across blades - - -	68 inches.
Revolutions - - - - - - -	480 per minute.
Steam flow - - - - - - -	1,200 lbs. per minute.
Actual steam volume - - - -	12.04 cubic feet per lb
Blade angle - - - - - - -	20° (normal blades).
Assume blade efficiency as - - -	74 per cent.
Blade heights - - - - - -	$1\frac{1}{2}$ inches.

$$\text{Then, } V_t = \frac{68 \times 3.1416 \times 480}{12 \times 60} = 142 \text{ feet per second.}$$

$$V_s = \frac{\text{cubic feet steam flow per second}}{\text{clear area through blades}} = \text{feet per second.}$$

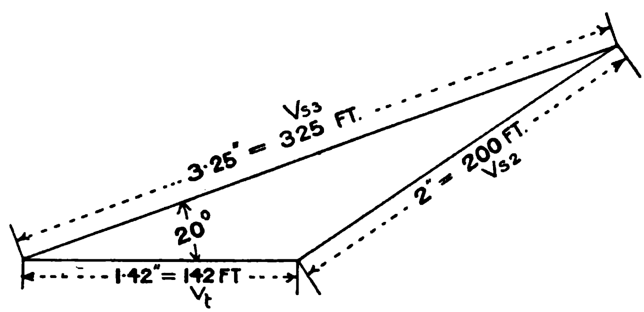
$$\text{Cubic feet flow per second} = \frac{1200 \times 12.04}{60} = 240.8 \text{ cubic feet.}$$

$$\text{Clear area between blades} = \frac{68 \times 3.1416 \times 1.5}{3 \times 144} = .741 \text{ square foot.}$$

Therefore, $V_s = 240.8 \div .741 = 325$ (nearly) feet per second.

NOTE.—For normal blades divide by 3 for clear area, and by 144 to obtain square feet.

$$\text{Then, } V_t = \frac{68 \times 3.1416 \times 480}{12 \times 60} = 142 \text{ feet per second.}$$



Velocity Triangle L.P. Turbine.

Set off the angle of 20° and measure up the exit steam velocity of 325 (3.25 inches) as shown, then set off 142 feet (1.42 inches) as the blade speed, and by completing the triangle the initial velocity of the steam measures 200 (2 inches nearly) feet per second.

$$\text{Work done at this expansion} = \frac{(325^2 - 200^2) \times 1200 \times 8 \times .74}{64.4}$$

$$= 7239130.43 \text{ foot-pounds.}$$

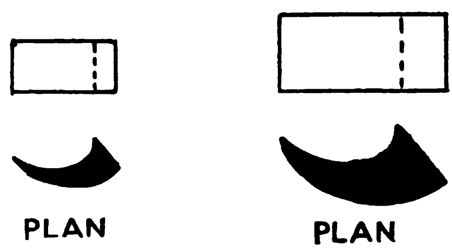
$$\text{Horse-power} = 7239130.43 \div 33000 = 219.3.$$

Number of Expansions.—The usual number arranged is 4 expansions in the H.P. turbine, and 8 expansions in each L.P. turbine, the total number of rows of blades being the same for each separate turbine. So that if the H.P. turbine is made up of, say, 4 expansions, each containing 16 rows of blades, then each L.P. turbine will have 8 expansions, each containing 8 rows of blades. Or again, suppose the H.P. turbine to consist of 4 expansions with 14 rows of blades in each, then the L.P. turbines will consist of 8 expansions with 7 rows of blades in each. It must of course be understood that the rotors will in this case contain 56 rows in all, and the casings 56

rows in all; or 56 rows of moving blades and 56 rows of fixed blades for each turbine, H.P. and L.P. A pair of rows, consisting of one row of fixed and one row of moving blades, is usually called a "stage." So that in the above case the H.P. turbine will have 14 stages in each expansion, and the L.P. turbines have 7 stages in each expansion.

The "packing pieces," which are caulked in between the blades, vary in shape according to position, being of small thickness section and much curved at the first expansions, and of larger thickness section and less curvature at the last expansions. The shape of the packing piece exactly corresponds to the two surfaces of the blades between which it fits in. So that there is a much smaller curve on the side in contact with the steam-receiving surface of the blade than that in contact with the convex surface or back of the blade.

Packing Pieces.—The small packing pieces which are caulked in between every pair of vanes are made of brass, and vary in size according to the position occupied by them in the turbine, being small



"**Packing Pieces**" (**full size**).

at the forward or higher pressure end, and larger at the after or lower pressure end (see sketch).

Effect of opening up of blades:—

1. With a limited steam flow per second the velocity of the steam will be reduced with loss of efficiency.

2. With unlimited steam supply the velocity may remain constant, but the quantity of steam flowing per second will then increase, thus developing more speed and power in the turbines.

In a case which recently came under the writer's notice, the turbines of a steamer did not develop the full power during trials; as a remedy the turbine covers were lifted, and a staff of men set to work to *open up* the exit edges of the blades of the first two or four rows in each L.P. expansion; this alteration had the desired effect of inducing a quicker steam flow per second, and in increasing the power of the turbines and the speed of the steamer to that required by the contract. Before the alteration the turbines did not take away the steam fast enough from the boilers, notwithstanding the fact that the stop-valves were full open. This occurrence is unusual. Before the alteration the L.P. turbine receivers indicated a high pressure,

In th

L.P.

but afterwards this pressure fell off to a normal figure. Obviously the L.P. turbines were unable to use up the steam, thus producing a high back pressure against the H.P. turbines, and resulting in reduced power in all the turbines.

"Wing" Blades.—At the 6th, 7th, and 8th L.P. expansions the blade heights are usually the same, but the 7th and 8th expansions are fitted with blades of much flatter section than the normal blades, and which give a much increased area of exit opening ; these blades are termed "wing" blades, and are made in three types, "semi-wing," "wing," and "double wing." The usual arrangement is to fit "semi-wing" and "wing" blades in the 7th and 8th L.P. expansions, but in the large Cunarders "double wing" blades were also fitted.

The "wing" type of blades give an area ranging from about 50 per cent. to 85 per cent. of the annulus area, which allows of moderate blade heights at the last L.P. expansions where the steam volumes to be handled may be anything from 40 to 100 cub. ft. per pound.

"Wing" blades are also fitted in the last reverse turbine expansions.

In blading lists as issued to the blading staff, the various blades are indicated by a three-figure number, the second figure of which represents the number of eighths in the blade widths, and the last figure the type of blade, whether "normal," "semi-wing," "wing," or "double wing."

Thus,	230	means	"normal"	$\frac{3}{8}$-inch	blades.
	240	,,	,,	$\frac{1}{2}$-inch	,,
	251	,,	"semi-wing"	$\frac{5}{8}$-inch	,,
	262	,,	"wing"	$\frac{3}{4}$-inch	,,
	263	,,	"double-wing"	,,	,,
	302	,,	"wing"	$1\frac{1}{4}$-inch	,,
	381	,,	"semi-wing"	1-inch	,,

Blade Speed and Steam Speed Ratio.—The ratio V_t to V_s at any expansion can be determined as follows :—

EXAMPLE.—Compare the ratio of V_t to V_s at the 1st H.P. expansion and at the 8th L.P. expansion, the steam flow being 31 lbs. per second through the H.P. turbine, and 15.5 lbs. per second in the L.P. turbine.

The revolutions are 470 per minute, the H.P. rotor drum 48 in. diameter, and the L.P. rotor drum 68 in. diameter, the H.P. 1st expansion blades $1\frac{1}{2}$ in. in height, and the L.P. 8th expansion blades 8 in. in height.

The actual steam volume per pound at the H.P. 1st expansion is 2.88 cub. ft., and at the 8th L.P. expansion 140 cub. ft. ; the clear exit area for steam flow is one quarter of the H.P. annulus area, and one half of the L.P. annulus area.

H.P. Turbine.

$$\text{Then, } V_t = \frac{\text{Mean blade circumference} \times \text{revolutions}}{60} =$$

$$\frac{(48 \text{ in.} + 1\tfrac{1}{2} \text{ in.}) \times 3.1416 \times 470}{12 \times 60} = 101.5 \text{ ft. per second.}$$

NOTE.—12 in. to reduce to feet.
NOTE.—60 seconds to reduce to seconds.

$$\text{Exit area} = \frac{\text{Mean blade circumference} \times \text{blade height}}{\text{clear area ratio}} =$$

$$\frac{(48 + 1.5) \times 3.1416 \times 1.5}{4 \times 144} = .404 \text{ sq. ft.}$$

NOTE.—144 sq. in. to reduce to sq. ft.

$$\text{Therefore, } V_s = \frac{\text{lbs. steam per second} \times \text{volume}}{\text{exit area}} = \frac{31 \times 2.88}{.404} = 220 \text{ ft. per sec.}$$

Then, Ratio V_t to $V_s = 101.5 \div 220 = .46$.

L.P. Turbine.

$$\text{Again, } V_t = \frac{(68 + 8) \times 3.1416 \times 470}{12 \times 60} = 155 \text{ ft. per second.}$$

$$\text{Exit area} = \frac{(68 + 8) \times 3.1416 \times 8}{2 \times 144} = 6.63 \text{ sq. ft.}$$

$$\text{Therefore, } V_s = \frac{15.5 \times 140}{6.63} = 327.$$

Then, Ratio V_t to $V_s = 155 \div 327 = .47$.

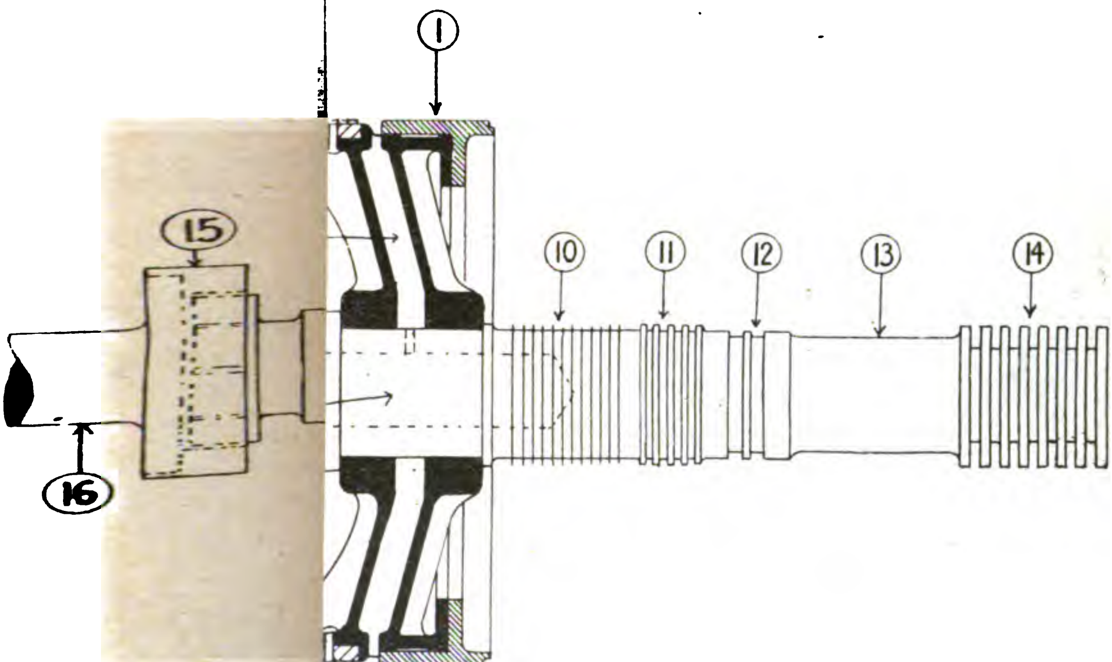
Cruising Turbines.—In Admiralty work cruising turbines are fitted to allow of low speeds and powers, together with fair economy. The cruising turbines are distinctive from the ordinary turbines in the following points:—

1. Longer rotor drums, owing to the greater number of blade rows required per expansion.
2. Smaller blade differences per expansion, this being necessary to keep down the difference of pressure and steam velocity, so that with the reduced mean blade speed required, the ratio of blade speed to steam speed will not be far from .43 to .48.
3. Smaller blade heights and rotor diameter due to smaller steam flow per second for the reduced H.P.

The number of expansions are usually either 3 or 4, and each expansion consists of from 16 rows to 22 rows of blades. In cruisers of the "Bellerophon"-"Boadicea" class the cruising turbines are fitted with blade heights as follows:—

1st expansion,	22 rows	of	blades,	$\frac{9}{16}$ in.	in height,	at	1 in.	pitch.
2nd	"	"	"	$\frac{11}{16}$ in.	"		1 in.	"
3rd	"	"	"	$\frac{15}{16}$ in.	"		$1\frac{1}{4}$ in.	"
4th	"	"	"	$1\frac{1}{4}$ in.	"		$1\frac{1}{4}$ in.	"

The drum diameter being 68 in. and the length 8 ft. 5 in.



(1) Dum
(2) First n spindle.
(3) Seco radial rings.
(4) Thir s for Ramsbottom rings.
e for baffle plate.

(13) Main bearing.
(14) Thrust.
(15) Sliding coupling.
(16) Main turbine shaft.

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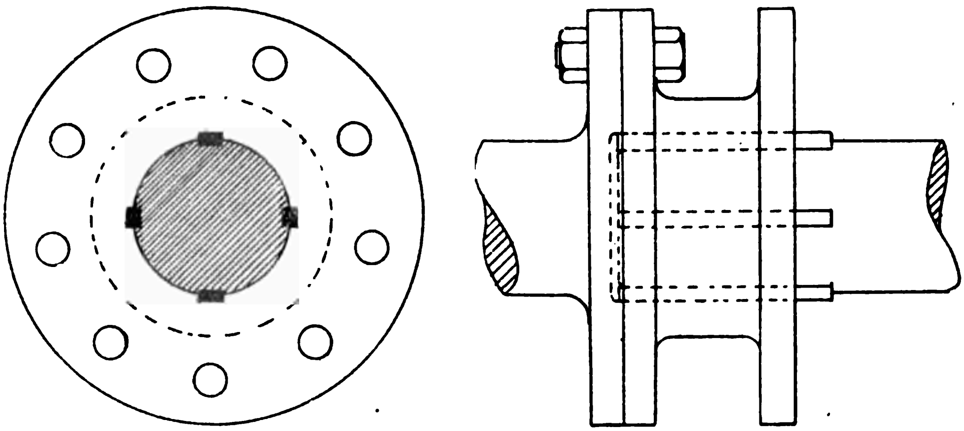
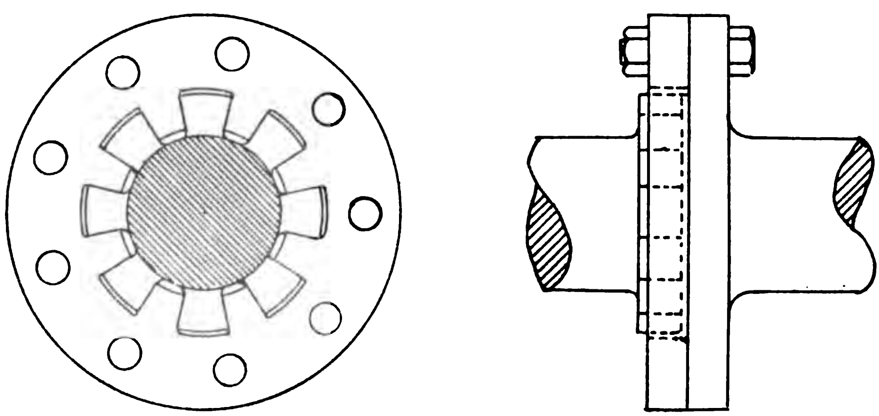


Sliding Coupling, with Loose Flange.



Sliding Coupling, with Teeth.

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The blade tip clearance (cold) for the above varies from $\frac{40}{1000}$ in. at the 1st expansion to $\frac{50}{1000}$ in. at the 4th or last expansion; when heated up, however, the actual blade tip clearance is only about two-thirds of the foregoing at the 1st expansion, and rather less at the last expansion, the brass blades expanding more than the steel rotor drum or the cast-iron casing. The cruising dummy clearance cold is only about $\frac{15}{1000}$ in., but generally, when heated up, this increases to about $\frac{25}{1000}$ in., or even more. The steam, after passing through the first cruising turbine, enters the second or M.P. cruising turbine, if one is fitted, then the H.P. ahead and L.P. ahead turbines, exhausting finally to the condenser. If only one cruising turbine is fitted to each set, the steam exhausts from it direct to the ahead H.P. turbine, which is the arrangement in the "Indomitable"-"Inflexible" class.

During the trials of the "Indomitable" and "Inflexible" speeds of over 20 knots were easily obtained with the cruising turbines connected up in series with the main turbines.

Adjustment and Expansion Allowance.—The cruising turbines being in line with the main turbines require some type of adjustable coupling for connecting the shafts of each, and the two systems which have been adopted in practice are:—

1. Bolted coupling, with loose or sliding flange on one of the shafts, moving longitudinally on feathers.
2. Flanges arranged to fit together by a system of teeth and recesses arranged circumferentially as shown in the sketch. This method is certainly the simplest, and is used in torpedo boat practice, and in vessels of the "Boadicea" class, although perhaps not so mechanically perfect as the other.

The sliding coupling is necessary to allow of separate "face" dummy clearance adjustment in the cruising turbine and in the ahead turbine, also to provide for independent expansion of each without one affecting the other. In large vessels the cruising turbines are generally fitted with the "face" type of dummy ring similar to the ahead turbines, whereas the astern turbines, whether H.P. and independent, (see "Indomitable"), or incorporated with the L.P., are fitted with the "fin" type of ring. The "fin" type of dummy gives large longitudinal clearance spaces without danger of contact, and thus allows of ample variation of adjustment of the ahead "face" ring dummy without risk of the astern dummy being damaged. This being the case, when H.P. astern turbines are fitted on the H.P. shaft, and independent of the H.P. turbine, a solid coupling is sufficient for connecting up the two turbines (as shown in the sketch), instead of a sliding coupling as required for connecting the cruising turbines.

Sometimes, however, in the case of smaller cruisers or battleships, a solid coupling is employed between the cruising and ahead turbines, and the cruising turbine dummies are then made of the "fin" type, similar to the astern, to allow of adjustment of the ahead turbine on the same shaft, and also for expansion. In this case the cruising

thrust-block receives a share of the propeller thrust, and the steam pressure thrust of the cruising blades acts as a counterbalance to the propeller thrust pressure.

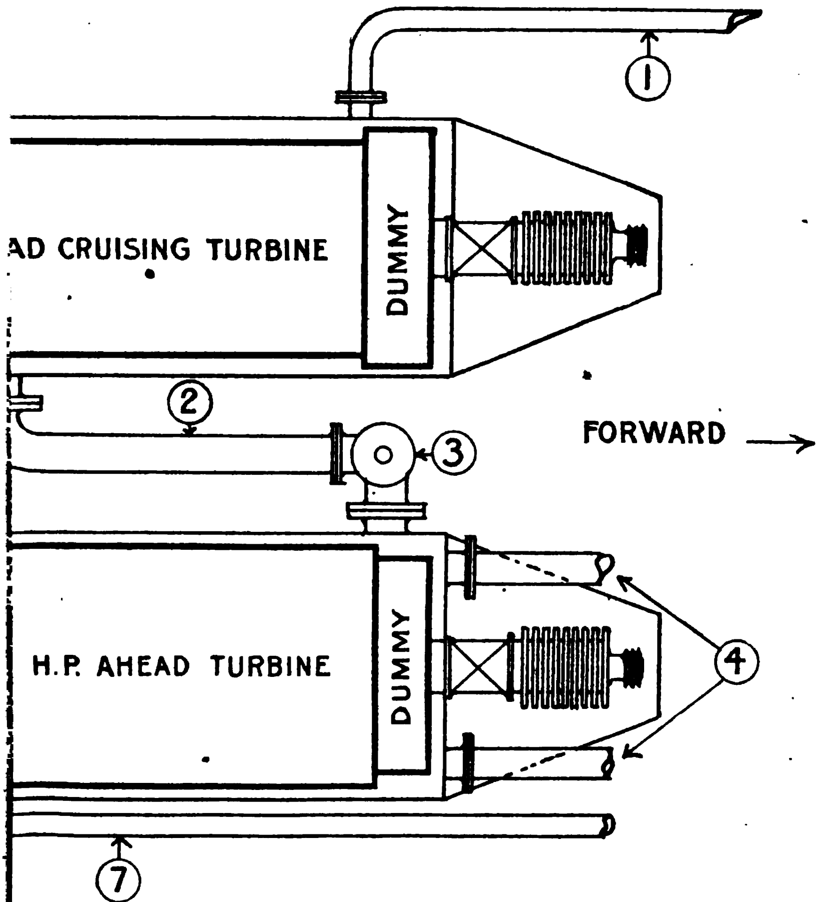
The "fin" tip clearance is usually $\frac{20}{1000}$ or .020.

Cutting Out of Cruising Turbines.—The engineering staff of a Government vessel recently made the interesting discovery that the consumption per horse-power at cruising speeds was much about the same when running with the main turbines only and the stop valves shut in as when running with the cruising turbines connected up and receiving boiler steam direct. This no doubt accounts for the recent decision of the Admiralty to, in future, discontinue the fitting of cruising turbines to battleships and cruisers. Again, experience has proved that the "fan resistance" of the cruising turbine blades when running ahead with the main turbines only resulted in a somewhat serious loss of power, and to partly overcome this, it was found advisable to ensure a vacuum in the cruising turbines *when cold.* This was done by increasing the fitting of a special "lantern" steam gland on the steam pipes, in addition to the usual expansion joint. When running with the main turbines only, it was found that the cruising turbines were drawing in air through the expansion joint of the steam pipes, which, although tight enough when heated up under steam, were allowing the admission of air when cold, owing probably to the contraction when cooling.

The air thus admitted increased the resistance opposed to the cruising blades when revolving idly, and absorbed a certain amount of power. By the fitting up of the "lantern" type of steam gland on the steam pipe outside of the ordinary expansion joint, the air is kept out, and a vacuum maintained inside the turbine casings, which reduced the resistance to be overcome by the moving blades, and which proportionally set free a certain amount of horse-power which could be effectively utilised for purposes of propulsion.

As will easily be imagined, the disconnecting of the cruising turbines altogether when running ahead would result in still further improvement, as all drag or resistance would then be taken off the ahead turbines. The same remark holds good as regards the astern turbines, which, whether independent or incorporated, act as a hindrance to ahead running. In like manner, when running astern, the ahead turbines act as a resistance, and represent so much lost power, which, if tested, would likely be found to be somewhat considerable. The above constitutes a rather important drawback to the turbine, as, owing to the existing conditions, full economy is impossible of attainment.

Cruising Thrusts.—The cruising turbines are fitted with small thrusts similar to those of the ahead turbines, but it should be noted that no propeller thrust is taken up by these, as the sliding coupling interposed between the ahead turbine in line with the corresponding cruising turbine prevents the cruising thrust block from receiving any



rn to L.P. astern turbine.
v of independent rotor adjustment.
tern fitted with radial fin rings).

umps.

g aft counterbalances the propeller thrust forward.
counterbalances the propeller pull aft when running astern.
counterbalance the propeller pull aft when running astern.
on the blades aft (no propeller thrust takes effect on this turbine owing to

e anchored aft and free to expand forward by means of oval holes for the [seating bolts at that end.

[To face page 48.

pressure from the propeller, the ahead turbine taking up its own thrust and that produced by the cruising turbine in addition. The cruising thrust block is required for two reasons :—

1. To allow of the rotor and dummy adjustment.

2. To take up the *steam* thrust on the blades, which is exerted from the initial to the exhaust end.

From the foregoing it will perhaps be obvious that, should the cruising turbines be worked at an overload, the ahead turbine thrust of that shaft may be in danger of overheating owing to the excessive thrust pressure set up, and which may not be balanced by the reduced pressure carried in the ahead turbine when running with the cruising turbine working. The ahead turbine thrust, if designed only for the full pressure carried in that turbine, may not have sufficient surface to take up the combined propeller thrust of its own and the cruising turbine, because the steam thrust on the ahead blades being less and the cruising turbine taking none of its own propeller thrust, nor yet giving any counterbalancing steam thrust aft on the shaft, the dangers of excess pressure will be evident.

Cruising Turbine Glands. — The cruising turbine glands are similar to those of the H.P. turbines, being fitted with a pocket to which steam is led when heating up, and which afterwards acts as a "leak-off" when running.

Cruising Turbine Drains.—The cruising turbine drains are led away to the "after" end of the L.P. turbines, and are thus in direct connection with the wet air pumps. When running, the drain valves at the exhaust end are only kept slightly open, and the drains at the initial end shut. When stopped, however, all the drains are opened up full to clear the turbines of condensed water. If the drains were full open when running, steam would, of course, be blown out with the water.

Regarding the foregoing, the naval practice is as follows :—

Previous to lighting fires.	Open screw-down drain valves at both ends of cruising turbines, ahead ends of L.P. turbines (main), and after end of H.P. main turbines.
With H.P. cruising turbines under steam.	All screw-down drains to be closed down, except drains at exhaust ends of cruising turbines which should be kept slightly eased.
With M.P. cruising turbines under steam (H.P. cruising shut off).	Drains on H.P. cruising turbines to be slightly eased.
With H.P. and M.P. cruising turbines shut off.	Drains on both to be slightly eased.

Steam Connections.—When the cruising turbines are arranged on the compound system boiler steam is led direct to the H.P. cruising, and exhausts from the H.P. to the I.P. cruising (on another shaft),

from there the steam exhausts into the H.P. ahead turbine, and L.P. ahead turbine, finally passing to the condenser. This method of working is arranged for at the lowest powers and speeds; for higher speeds the boiler steam is led direct to the I.P. cruising turbine, then the H.P. ahead, L.P. ahead, and condenser. For full powers and speeds the cruising turbines are cut out altogether, and the boiler steam led direct to the H.P. turbine, then L.P. and condenser.

The cruising turbines are only fitted for ahead running. In the "Indomitable"-"Inflexible" class of vessel with double compound ahead turbines, two cruising turbines are fitted, both being H.P., so that when running at low speeds and powers the boiler steam is led direct to each cruising turbine, and from there to the corresponding H.P. ahead and L.P. ahead turbines and so to the condenser of that set.

Reverse Turbines.—The reverse turbines consist of a set of smaller blades placed in the after end of the L.P. turbine casings. The angle of these blades is the same as the ahead, but the curve is on the other side, and as the steam is admitted at the *opposite* end of the turbine casing, the direction of shaft rotation is reversed. Both the ahead and reverse turbines exhaust into the same central exhaust passage leading to the condenser. The reverse turbines are also fitted with dummy pistons, fitted with "radial fin" type of rings.

The reverse casing is fitted with small drain holes (in the lower half), one hole for each expansion (at the end row of each). The water so drained off passes into the exhaust space, and then into the condenser.

In the latest types of turbine steamers the size of the reverse turbines has been considerably increased.

The reverse rotor is often made of the same diameter as the H.P. rotor, and the reverse power developed equal to about one-half or even more of the ahead power.

H.P. and L.P. Astern Turbines.—In naval practice astern turbines are often fitted to each shaft, and if four lines of shafting, these are arranged as H.P. and L.P. astern, two of each in all, the H.P. asterns being placed on the ahead H.P. turbine shafts, and the L.P. asterns on the ahead L.P. turbine shafts.

In running astern, steam is led first to the H.P. astern turbines from which it exhausts to the L.P. astern turbine, and from there to the condenser as usual.

The H.P. astern turbines are independent of the H.P. ahead turbines, but the L.P. astern turbines are incorporated with the ahead L.P. turbines. The initial end of the H.P. astern is *forward*, and the initial end of the L.P. astern *aft*, which arrangement requires a difference in the dummy diameter of each, the H.P. astern dummy being in excess of the corresponding rotor drum diameter, and the L.P. astern dummy being less than the corresponding rotor drum diameter. This allows of an annulus, on which the steam pressure will, in both H.P. and L.P. asterns, act *forward* and thus balance to a certain degree the propeller pull aft when running astern.

The following show the respective diameters of rotor drum and dummy for ahead, astern, and cruising turbines :—

Turbines.	Drum Diameters.	Dummy Diameters.
H.P. ahead -	90 inches.	90 inches.
H.P. astern -	80 ,,	83 ,,
L.P. ahead -	114 ,,	96 ,,
L.P. astern -	76 ,,	72 ,,
Cruising - -	90 ,,	92 ,,

In the ahead and cruising turbines the dummy diameter is usually less than the drum, and the steam pressure acting on the small annulus so provided counterbalances (at certain speeds) the propeller thrust forward.

Trailing Shaft.—Collars are fitted in the length of tunnel shafting next the propeller shaft, on each side of the bearing, to act as a check on the longitudinal movement of the shafting, in the event of, say, a broken propeller or broken tail-end shaft. Should the propeller shaft break the loss of thrust in a forward direction might result in damage to the turbine by the steam thrust on the turbine blades aft; the check collar described will then bear up against the end of the tunnel bearing, and thus limit the end play of the shafting. A suitable clearance (from ½ in. to ¾ in. when cold), to allow of expansion when heated up under running conditions is allowed between the collar and faced-up end of the block, which is either of brass or of white metal.

Removable Thrust Rings.—When no adjustable coupling is fitted between the main and cruising turbines, the cruising thrust is fitted with loose horse-shoe type brass rings, which require to be taken out when running with the H.P. and L.P. turbines only. This is necessary owing to the danger of the unequal expansion between the main and cruising turbines "binding" the shaft when running ahead with the main turbines heated up and the cruising turbines cold.

Horse-Power of Turbines.

Shaft or Brake Horse-Power.—It is now well known that so far no method has been devised, or, in fact, is likely to be devised, for the indicating of the horse-power as done in the case of reciprocating engines, but the actual power transmitted along the shafting to the propeller may be determined by means of the "torsion meter" (pages 255-262), an instrument which measures the twist or torque put on the shaft by a given power. For accuracy of results it is advisable to have the shafting calibrated (page 133) beforehand, as different builds of shafts and materials give slightly varying results.

It should be noted that the shaft horse-power or brake horse-

power, as measured by the torsion meter, is the useful horse-power, and that the I.H.P. by comparison is a matter of indifference, the effective horse-power being actually transmitted along the shafting to the propeller being of chief importance.

Turbine Horse-Power.—The turbine horse-power may be found by calculation if the steam consumption, heat drop per pound steam, and turbine over-all efficiency are known, thus—

$$\text{Turbine Horse-Power} = \frac{\text{Heat drop} \times \text{Pounds steam per min.} \times 778 \times \text{Efficiency}}{33000}.$$

EXAMPLE.—The calculated heat drop per pound of steam is 290 heat units, and the steam flow is 1,300 lbs. per minute, determine the turbine horse-power if the turbine over-all efficiency is rated as 62 per cent.

$$\text{Then, Horse-Power} = \frac{290 \times 1300 \times 778 \times .62}{33000} = 5510.$$

Equivalent I.H.P.—Repeated trials have proved that the ratio of shaft horse-power by torsion meter as compared to indicated horse-power is usually in the ratio of 90 to 100, or .9 to 1.

Therefore, Equivalent I.H.P. = Shaft Horse-Power ÷ .9.

EXAMPLE.—The collective shaft horse-power by torsion meter is found to be 8100; calculate the equivalent I.H.P.

Then, equivalent I.H.P. = 8100 ÷ .9 = 9000.

Turbine Diagram.—Heat drop and Pressure drop.

The diagram shown illustrates the successive variations in steam pressure and volume which occur throughout the H.P. and L.P. turbines. The figures given are, however, only approximate, and merely serve to indicate to the student the work done by the steam, together with the corresponding changes in pressure, volume, &c., in its passage through the turbine casings.

DATA FOR DIAGRAM.

Coal per hour	7 tons.
Evaporation	8.5 lbs.
H.P. turbine initial pressure	140 lbs. gauge (dryness 1).
L.P. turbines " "	20 " " (" .92).
Condensers	27 in. vacuum (" .78).
Efficiency (assumed)	60 per cent.

NOTE.—Adiabatic expansion assumed.

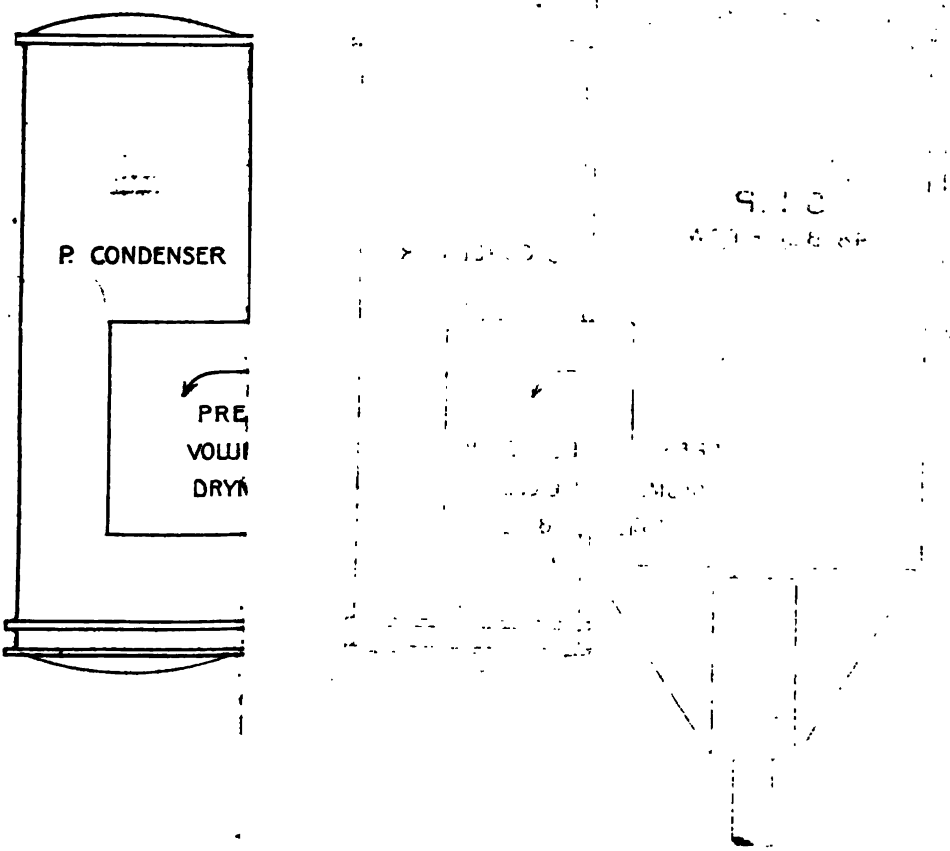
NOTE.—The H.P. terminal pressure is approximately the L.P. initial pressure, and the L.P. terminal pressures the condenser pressures.

H.P. Turbines.

$$\text{Steam flow per min.} = \frac{7 \times 2240 \times 8.5}{60} = 2223 \text{ lbs.}$$

Assuming leakage loss as 10 per cent.—

$$\text{Then, Actual steam flow through turbines} = \frac{2222 \times 90}{100} = 1998 \Big\} \text{(Say, 2000 lbs. per min.)}$$



Dis

Initial, $140 + 15 = 155$ lbs. absolute,

and, 155 lbs. = 361.1 temperature and 859.6 units latent heat (from Table, page 7).

$361.1 + 461 = 822.1$ absolute temperature.

Terminal, $20 + 15 = 35$ lbs. absolute.

and, 35 lbs. = 259.3 temperature and 931.6 units latent heat.

$259.3 + 461 = 720.3$ absolute temperature.

Heat drop per pound $= 859.6 \times 1 - 931.6 \times .92 + 822.1 - 720.3 = 104.33$ B.T.U.

$$\text{Horse-power} = \frac{104.33 \times 2000 \times 778 \times .60}{33000} = 2951.5.$$

Mean heat drop per blade row per pound steam $= 104.33 \div 48 = 2.17$ B.T.U.

Total pressure drop $= 155 - 35 = 120$ lbs.

Mean pressure drop per blade row $= 120 \div 48 = 2.5$ lbs.

L.P. Turbines.

Initial, 35 lbs. = 720.3 absolute temperature and 931.6 latent heat.

Terminal, 1.5 lbs. (absolute), 115.9 temperature, and 1033.2 units latent heat.

$115.9 + 461 = 576.9$ absolute temperature.

Heat drop per pound $= 931.6 \times .92 - 1033.2 \times .78 + 720.3 - 576.9$
$= 194.58$ B.T.U.

$$\text{Horse-power (each L.P.)} = \frac{194.58 \times 1000 \times 778 \times .60}{33000} = 2752.4.$$

NOTE.—Each L.P. turbine receives *half* the total steam flow, or say 1,000 lbs. per minute.

Mean heat drop per blade row $= 194.58 \div 48 = 4.05$ B.T.U.

Total pressure drop $= 35 - 1.5 = 35.5$ lbs.

Mean pressure drop per blade row $= 33.5 \div 48 = .7$ lbs. (nearly).

Total horse-power $= 2951.5 + 2752.4 + 2752.4$
$= 8456.3$ (say 8456).

$$\text{Steam per hour per horse-power} = \frac{2000 \times 60}{8456} = 14.19 \text{ lbs.}$$

Coal per hour per horse-power $= 14.19 \div 8.5 = 1.6$.

NOTE.—Referring to Steam Table (page 3) the specific volume at 35 lbs. pressure absolute is 11.64 cub. ft., and at 1.5 lbs. absolute, 225.5 cub. ft., and these multiplied by the respective dryness fractions obtain the actual steam volumes, thus :—

$11.64 \times .92 = 10.7$ cubic feet.
$225.5 \times .78 = 176$ „ „

Blade Tip Leakage Area.—The leakage over blade tips depends on the clearance between the rotor blades and the inside of the casing, and the clearance between the casing blades and the outside of the rotor. The so-called "blade height" includes the two working clearances referred to, and when these are deducted the effective heights of the blades are correspondingly reduced. To take an example :—

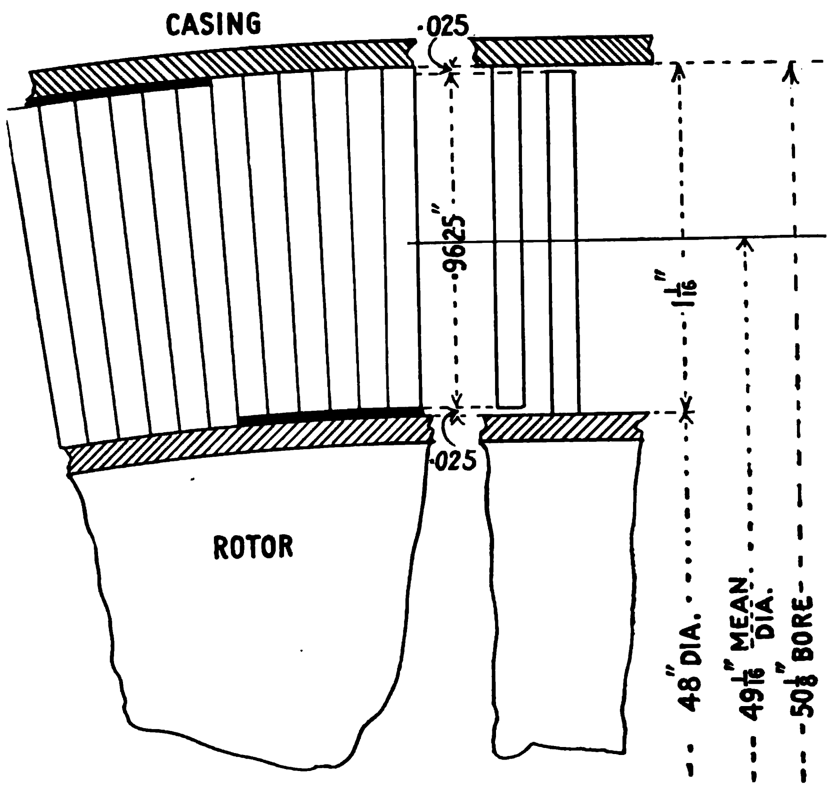
Rotor diameter outside, 48 in.
Blade height at 1st expansion, $1\frac{1}{16}$ in.
Tip clearance, $\frac{25}{1000}$ in., or .025 in.

From the foregoing it will be obvious that the effective blade height is equal to $1\frac{1}{16}$ in. $-$ .025 in. $\times$ 2 in. $=$ 1.0125 in., and the *mean* diameter of blades 48 in. $+ 1\frac{1}{16}$ in. $= 49\frac{1}{16}$ in. To find the per cent. leakage over blade tips, if the blade openings at exit edge are equal to .3 of annular area formed by blades, then—

$$49.0625 \times 3.1416 \times .9825 \times .3 = 45.4 \text{ sq. in. clear area,}$$
$$\text{and } 49.0625 \times 3.1416 \times .025 \times 2 = 7.706 \text{ sq. in. leakage area.}$$

$$\text{Therefore, } \frac{7.706 \times 100}{45.4} = 16.9 \text{ per cent.}$$

NOTE.—$49\frac{1}{16}$ in. $=$ 49.0625 in.



Tip Clearance Areas.

NOTE.—The Black Section represents area open for leakage.

The tip clearance area is therefore equal to about 17 per cent. of the steam area through blades.

Number of Rows of Blades and Blade Heights.—The number of rows of blades, according to the formula given by Mr E. M. Speakman on page 23, works out as follows :—

EXAMPLE 1.—Calculate the required number of rows of blades of a turbine if the H.P. blade velocity is to be 100 feet per second.

RULE.—$1500000 = V_t^2 \times$ number of rows.

Therefore, $\frac{1500000}{V_t^2} =$ number of rows, and $\frac{1500000}{100^2} = 150$ rows of blades.

NOTE.—By this is meant that if the H.P. turbine develops the total power, 150 rows of blades will be required in the rotor and on the casing of that turbine, but as the H.P. develops only about one-third of the total power (with the standard 3 ahead turbine arrangement) then, $150 \div 3 = 50$ rows of blades for the H.P. turbine actually required. The same statement holds good for the L.P. turbines, only as the two L.P. turbines develop two-thirds of the total power, the result must be multiplied by this fraction.

EXAMPLE 2.—Calculate the required number of blade rows required in each L.P. turbine referring to Example No. 1, the L.P. blade velocity being 140 feet per second.

Then, $\frac{1500000}{140^2} = 76$, and $76 \times \frac{2}{3} = 50$ blade rows for each L.P. turbine.

EXAMPLE 3.—The total number of rows of blades in the turbines of a channel steamer is 192. Calculate the mean velocity (V_t) of the blades.

$$\text{Then, } 1500000 = V_t^2 \times \text{number of rows.}$$

$$\text{Therefore, } 1500000 = V_t^2 \times 192,$$

$$\text{and } \frac{1500000}{192} = V_t^2 = \frac{1500000}{192} = 7812;$$

$$\text{so that, } \sqrt{7812} = 88 \text{ feet per second} = V_t.$$

By the rule given, determine the number of blade rows required in the H.P. and L.P. turbines of the "Lusitania" and "Mauretania," given that the rotor drum diameters are 96 in. and 140 in., the revolutions 180 per minute, and the constant adopted 1400000.

Then, surface velocity of H.P. drum per sec. $= \frac{96 \times 3.1416 \times 180}{12 \text{ in.} \times 60 \text{ min.}} = 75.4$ ft.,

and, $\left.\begin{array}{l}\text{number of blade rows required}\\ \text{for full work of steam}\end{array}\right\} = \frac{1400000}{75.4^2} = 246$ rows of blades.

$\left.\begin{array}{l}\text{As, however, the H.P. is designed for only}\\ \text{half the work of the steam passing}\\ \text{through it, the L.P. to develop the}\\ \text{other half,}\end{array}\right\}$ then, $246 \div 2 = 123$ rows of blades actually required.

Again, surface velocity of L.P. drum per sec. $= \frac{140 \times 3.1416 \times 180}{12 \text{ in.} \times 60 \text{ min.}} = 110$ ft.,

and, $\frac{1400000}{110^2} = 115$ blade rows for full work.

Therefore, $115 \div 2 = 57$ blade rows for half work.

NOTE.—The actual blade rows fitted are: H.P. 123, and L.P. 56, which agree closely with the foregoing.

Initial Steam Velocity.—According to an article in *Engineering* of 13th December 1907, on "The Practical Proportioning of the Reaction Steam Turbine," the velocity of the H.P. turbine initial steam is determined by the following:—

$$\text{Rule, Velocity of steam at 1st H.P. expansion} = \frac{\text{Constant}}{\sqrt{N}}$$

Where Constant = 2700 (64 steam expansions assumed).
" " = 3000 (more than 64 expansions assumed).
" N = No. of blade rows in turbines from rule given on page 23.

The successive blade height ratios is given as $\sqrt{2}$, and the approximate ratio of H.P. to L.P. rotor drum also as $\sqrt{2}$, but these ratios vary somewhat in different cases, as may be tested by working out and comparing with the examples from actual practice, pages 138 to 218. To apply the foregoing to turbines as actually constructed, take the following cases:—

EXAMPLE 1.—No. of blade rows 166, and constant 2,700 for 64 expansions.

$$\text{Then, } \frac{2700}{\sqrt{166}} = \frac{270.0}{12.88} = 210 \text{ ft. (nearly) per second.}$$

From this the required clear area through blades, and therefore blade heights, may be calculated, given the steam flow per second, the actual volume of the steam, and the rotor drum diameter.

EXAMPLE 2.—Calculate the initial velocity of the steam at the H.P. turbine of the U.S.S. "Chester," given that the blade rows = 140 and the constant adopted = 2,700.

$$\text{Then, velocity} = 2700 \div \sqrt{140} = 2700 \div 11.83 = 228 \text{ ft. per second.}$$

EXAMPLE 1.—Determine the required blade height of the 1st L.P. expansion of the "Dreadnought" turbines, given that the rotor drum diameters are 68 in. H.P., and 92 in. L.P., the last H.P. expansion blades being 5 in. and the blade height increase, to allow of steam expansion, to be in the usual ratio of $\sqrt{2}$ times.

Then, Ratio of rotor diameters = $92 \div 68 = 1.35$.

For similar rotor diameters the 1st L.P. expansion blade height would now require to be equal to $5 \times \sqrt{2} = 7.05$ in. But as the L.P. rotor diameter is 1.35 times that of the H.P. rotor, then, $7.05 \div 1.35 = 5.22$ in. height of blade, neglecting increase of steam volume and velocity, which is also proportional to the increase of rotor diameters; therefore $5.22 \div 1.35 = 3.8$ in. or $3\frac{7}{8}$ in. blade height, which is the actual size adopted.

EXAMPLE 2.—Calculate the required blade height of the 1st L.P. expansion, given that the rotor drum diameters are 30 in. H.P. and 45 in. L.P., and the last H.P. expansion blade height $4\frac{1}{2}$ in. The turbines are three in number, one H.P. and two L.P.

Then, $4.5 \times \sqrt{2} = 6.36$ in. blade height for similar diameter of rotor.
and, ratio of rotor diameters $= 45 \div 30 = 1.5$.
Therefore, $6.36 \div (1.5 \times 1.5) = 2.82$ in. blade height if one L.P. turbine only is fitted, so that, $2.82 \div 2 = 1.41$ (say 1.5 in.) blade height for each L.P. turbine.

EXAMPLE 3.—Calculate the blade heights at each expansion, given the following:—Horse-power, 10,000; initial pressure, 140 lbs. by gauge (volume, 2.87 cub. ft.); steam consumption (estimated), 14 lbs. per horse-power hour; H.P. rotor drum, 48 in. diameter; L.P. rotor drum, 68 in. diameter; revolutions (designed), 500 per minute; blade rows 144, and constant adopted for initial steam velocity, 2,700. Assume normal blades giving a clear area ratio of one-third blade annulus.

H.P. Turbine.

$$\text{Initial velocity of steam} = 2700 \div \sqrt{144} = 225 \text{ ft. per second.}$$

$$\text{Steam flow per second} = \frac{10000 \times 14 \times 2.87}{60 \times 60} = 111.61 \text{ cub. ft.}$$

$$\text{Required clear area} = 111.61 \div 225 = .496 \text{ sq. ft.}$$

$$\text{Required annulus area} = .496 \times 3 = 1.488 \text{ sq. ft.,}$$

$$\text{and, } 1.488 \times 144 = 214.272 \text{ sq. in.}$$

$$\text{Therefore, diameter across blade tips} = \sqrt{\frac{214.272 + 48^2 \times .7854}{.7854}} = 50{\cdot}76,$$

$$\text{and, } \frac{50.76 - 48}{2} = 1.38 \text{ blade height at 1st expansion.}$$

EXAMPLE.—To find the blade height at the 2nd, 3rd, and 4th expansions—

Then, $1.38 \times 1.42 = 1.95$ in.,	say 2 in.	at 2nd expansion.
$2 \times 1.42 = 2.84$ in.,	,, $2\frac{3}{4}$ in.	,, 3rd ,,
$2.75 \times 1.42 = 3.9$ in.,	,, 4 in.	,, 4th ,,

NOTE.—Assuming that $\sqrt{2} = 1.42$.

L.P. Turbine.

$$\text{Drum ratio} = 68 \div 48 = 1.4166.$$

$$\text{Therefore blade height of 1st expansion} = \frac{4 \times 1.42}{1.4166 \times 1.4166 \times 2} = 1.4, \text{ say } 1\tfrac{3}{8} \text{ in.}$$

Blade heights at the other expansions—

1.375 in. $\times 1.42 = 1.95$ in.,	say 2 in.	at 2nd expansion.
2 in. $\times 1.42 = 2.84$ in.,	,, $2\frac{3}{4}$ in.	,, 3rd ,,
2.75 in. $\times 1.42 = 3.9$ in.,	,, 4 in.	,, 4th ,,
4 in. $\times 1.42 = 5.68$ in.,	,, $5\frac{1}{2}$ in.	,, 5th ,,
5.5 in. $\times 1.42 = 7.8$ in.,	,, 8 in.	,, 6th ,,
"semi-wing" blades	8 in.	,, 7th ,,
,, ,,	,, 8 in.	,, 8th ,,

NOTE.—The blade heights as shown are measured from *above* the surface of the drum only, the actual blade height over all being in excess of these by the depth of the fitting groove, which varies from $\frac{1}{4}$ in. in small H.P. blades to $\frac{3}{8}$ in. or $\frac{1}{2}$ in. in large L.P. blades. It should also be noted that the tip clearance is neglected in the above dimensions, which if deducted would reduce the blade heights by about $\frac{30}{1000}$ in. in the H.P. blades and about $\frac{50}{1000}$ in. in the L.P. blades.

In the "Lusitania" and "Mauretania" the increase of blade height is in the ratio of 1.22 (full) in the H.P. turbines and 1.27 (full) in the L.P. turbines; whereas in the "Invincible" class the ratios are 1.33 for H.P. blades and about 1.4 (full) for L.P. blades. The rotor drum ratio of the Cunarders is as 1 : 1.45 (full) and of the "Invincible" class as 1 : 1.25.

EXAMPLE 1.—Given that the blade height at the 1st H.P. expansion of the "Lusitania" is $2\frac{3}{4}$ in., calculate the blade heights of the 2nd, 3rd, 4th, 5th, 6th, 7th, and 8th H.P. expansions.

Then,	$2.75 \times 1.22 = 2.355$,	say $3\frac{3}{8}$ in.	at 2nd expansion.
	$2.375 \times 1.22 = 4.117$,	„ $4\frac{1}{4}$ in.	„ 3rd „
	$4.25 \times 1.22 = 5.185$,	„ $5\frac{1}{4}$ in.	„ 4th „
	$5.25 \times 1.22 = 6.405$,	„ $6\frac{1}{2}$ in.	„ 5th „
	$6.5 \times 1.22 = 7.930$,	„ 8 in.	„ 6th „
	$8 \times 1.22 = 9.76$,	„ 10 in.	„ 7th „
	$10 \times 1.22 = 12.20$,	„ $12\frac{1}{4}$ in.	„ 8th „

EXAMPLE 2.—Given that the 1st expansion blade height is $8\frac{1}{4}$ in., calculate in the same manner the blade heights at the 2nd, 3rd, 4th, and 5th L.P. turbine expansions, the remaining expansions (6th, 7th, and 8th) being of the special "winged" type.

Then,	$8.25 \times 1.27 = 10.477$,	say $10\frac{1}{2}$ in.	at 2nd expansion.
	$10.5 \times 1.27 = 13.335$,	„ $13\frac{1}{2}$ in.	„ 3rd „
	$13.5 \times 1.27 = 17.145$,	„ $17\frac{1}{4}$ in.	„ 4th „
	$17.25 \times 1.27 = 21.907$,	„ 22 in.	„ 5th „

NOTE.—This height of 22 in. also holds good for the 6th, 7th, and 8th expansions, but the blade openings are increased much in excess of the normal amount of .33 of the annulus area by these blades being of "semi-wing," "wing," and "double-wing" section and angles, which, in the case of the last named, almost brings the blades parallel to the drum in a fore-and-aft line.

EXAMPLE.—Calculate the blade heights of the 1st H.P. expansion of the "Indomitable," given that the drum is 92 in. diameter, the revolutions 250 per minute, the horse-power 40,000, and assuming 14.2 lbs. of steam per horse-power hour. Assume an initial pressure of 140 lbs. (gauge), and take number of rows as 120, with a constant of 2,700.

Then, Velocity of initial steam flow per sec. $= \frac{2700}{\sqrt{120}} = \frac{2700}{11} = 245$ ft.,

and, Steam flow per sec. $= \frac{20000 \times 14.2 \times 2.87}{60 \times 60} = 226.4$ cub. ft.

NOTE.—20,000 horse-power for each pair of H.P. and L.P. turbines.

" 140 lbs. gauge = 155 lbs. absolute = 2.87 cub. ft. specific volume. See "Steam Table," page 3.

Clear area required = $226.4 \div 245 = .924$ sq. ft.

Annulus area required = $.924 \times 144 \times 3 = 399$ sq. in.

Therefore, Diam. across blade tips $= \sqrt{\frac{399 + (92^2 \times .7854)}{.7854}} = 94.72$ in.,

and, Blade heights $= \frac{94.72 - 92}{2} = 1.36$ in., say $1\frac{3}{8}$ in.

EXAMPLE.—Also proceed to determine the blade heights of the 2nd, 3rd, 4th, 5th, and 6th H.P. expansions of the same turbine, the increase of height being in the ratio of 1.33 (full).

Then, $1.375 \times 1.33 = 1.828$, say $1\frac{7}{8}$ in. at 2nd expansion.
$1.875 \times 1.33 = 2.493$, " $2\frac{1}{2}$ in. " 3rd "
$2.5 \times 1.33 = 3.325$, " $3\frac{1}{2}$ in. " 4th "
$3.5 \times 1.33 = 4.655$, " $4\frac{3}{4}$ in. " 5th "
$4.75 \times 1.33 = 6.317$, " $6\frac{3}{8}$ in. " 6th "

NOTE.—In the foregoing the number of blades in each expansion will probably vary from about 16,000 in the 1st to about 9,000 in the 6th of the rotor, and from about 11,000 in the 1st to about 7,000 in the 6th of the casing.

EXAMPLE.—Calculate (1) the area open for steam flow; (2) the steam flow in cubic feet per second; (3) in lbs. per second; and (4) the horse-power developed, allowing 15 lbs. steam per horse-power hour, for the following ("Lusitania"):—The H.P. rotor drum being 96 in. diameter; blade height at 1st expansion, $2\frac{3}{4}$ in.; blade opening at exit edges, .09 in.; revolutions, 180 per minute; initial pressure, 180 lbs. gauge; and assuming a mean blade speed to steam speed ratio of .42 at this expansion. Assume the number of blades per row to be 1000, and neglect blade tip clearance.

Then, Mean diameter across blades = 96 in. + 2.75 in. = 98.75.

Mean blade speed in ft. per second $= \frac{98.75 \times 3.1416 \times 180}{12 \times 60} = 77.5$ ft.

Steam speed per second $= 77.5 \div .42 = 184$ ft.

Area of exit opening between each pair of blades $= .09 \times 2.75 = .2475$ sq. in.

" " " per ring of blades $= \frac{.2475 \times 1000}{144} = 1.718$ sq. ft.

Cub. ft. steam flow per second $= 1.718 \times 184 = 316.11$ cub. ft.

Pounds steam flow " " $= 316.11 \times .432 = 136.55$ lbs.

" " " hour $= 136.55 \times 60 \times 60 = 491580$ lbs.

Horse-power (one side) $= 491580 \div 15 = 32772$.

Horse-power (both sides) $= 32772 \times 2 = 65544$.

NOTE.—185 lbs. + 15 = 195 lbs. absolute, and 195 absolute = .432 density, from "Steam Table," page 3.

NOTE.—The actual area of opening as given between blades for steam flow is less than the third usually assumed for normal blocks, as the following calculations prove:—

$$\text{Blade annulus (clearance neglected)} = \frac{(96 + 2.75) \times 3.1416 \times 2.75}{144} = 5.92 \text{ sq. ft.}$$

Ratios of actual opening to annulus $= 1.718 \div 5.92 = .2902$.

The openings between blades can be varied during construction by means of "closing up" and "opening out" tools for the purpose which change the blade exit angle as considered advisable. This work is occasionally done after trials, and in some cases has produced decided improvement in the results (see page 44). It might be argued from this that a certain amount of "trial and error" method is still required in turbine work to obtain the best results.

Diameters of Rotors and Dummies.—The following figures give the respective sizes of the rotor drums and dummies as usually designed for the turbines of channel steamers:—

Turbine.	Diameter of Rotor Drum.	Diameter of Dummy.
H.P. - -	5 feet 3 inches.	5 feet 0 inches.
L.P. - -	7 " 6 "	6 " 8 "
Reverse -	5 " 10 "	5 " 9 "

NOTE.—The steam pressure acting on the annulus area formed by the difference of drum and dummy diameter together with the effective combined steam pressure on the blades is designed to balance the propeller speed at maximum speed, although at lower speeds the propeller thrust may either more or less than balance this pressure.

Water Condensed in Turbines.—Starting with dry saturated steam at the initial end of the H.P. turbine, a mixture of condensed steam and water is produced at the last L.P. expansion due to the (partial) adiabatic expansion of the steam in doing work. This must not be confused with the initial condensation which occurs in reciprocating engine cylinders, and which is due to the alternate heating up and cooling down during admission and exhaust.

A small amount of condensation takes place in the H.P. turbine due to the decrease of dryness fraction, but much more occurs in the L.P. turbines, which accounts for the "wet" air pump connection to the after end of these turbines.

According to the "error" diagram of Mr E. M. Speakman (page 21), the actual volume of the steam at a pressure of 2 lbs. absolute is

143.4 cub. ft. per lb., which gives a dryness fraction of .833. If then the steam flow be known, the condensed water produced can be found as follows :—

Water condensed in turbines per minute = steam flow per minute × (1 − .833) (assuming a terminal L.P. pressure of 2 lbs. absolute).

EXAMPLE.—H.P. initial pressure, 150 lbs. gauge; H.P. terminal pressure, 22 lbs. gauge; and L.P. initial pressure, 20 lbs. gauge.

Horse-power, 16,000, and assume 14 lbs. steam per horse-power hour. Assume H.P. initial steam of dryness 1.

According to the "error" diagram of Mr Speakman, 35 lbs. pressure absolute = .932 dryness factor and 2 lbs. pressure = .835 dryness factor (page 41).

NOTE.—20 + 15 = 35 lbs. absolute.

Then, total weight of water condensed in turbines $= 16000 \times 14 \times (1 - .835) = 36960$ lbs. per hour.

Again, weight of water condensed in H.P. turbine $= 16000 \times 14 \times (1 - .932) = 15232$ " "

Weight of water condensed in each L.P. turbine $= \frac{16000 \times 14}{2} \times (.932 - .835) = 10864$ lbs. "

As mentioned elsewhere, most of the water condensed in the H.P. turbine is blown out with the steam into the two L.P. turbines, from which it is drawn off by the wet air pumps. These pumps are kept working when the turbines are stopped temporarily, in addition to working continuously when running.

Power Developed by each Turbine.—(*A.*) Referring to Example No. 16, page 173, it will be seen that, running at a trial speed of 14.73 knots, the pressures indicated were as follows :—

H.P. turbine - - - - - -	35 lbs. (gauge).
L.P. turbines - - - - - -	12 in. vacuum.
Condenser - - - - - - -	27 in. "

Assuming that the power varies as the cube of the speed, and as 9,000 horse-power gives 22 knots,

$$\text{Then, } \frac{14.73^3 \times 9000 \text{ I.H.P.}}{22^3} = 2701 \text{ Horse-Power.}$$

As there are three shafts, $2701 \div 3 = 900$ Horse-Power developed by each turbine; or, $900 \times 33,000 = 29,700,000$ foot-pounds of effective work per minute.

This energy is given out by each L.P. turbine in working between an initial pressure of 9 lbs. absolute and final pressure of about 2 lbs. absolute, or $9 - 2 = 7$ lbs. pressure drop in all, and this total pressure drop of 7 lbs. gives a heat drop sufficient to obtain 900 horse-power.

As, 29700000 foot-pounds $\div$ 778 = 38174 Units Heat Drop per minute.

If each L.P. turbine is made up of, say, 64 rows of blades in all, then the mean pressure drop per row is equal to .10 of a pound, as

$7 \div 64 = .10$; but the actual pressure drop is of a gradually decreasing quantity from the initial to the exhaust end of each turbine.

(*B.*) Again, taking the lowest trial speed of 10.68 knots, we find that the power required amounts to

$$\frac{10.68^3 \times 9000}{22^3} = 1030 \text{ Horse-Power.}$$

And as this is developed by the three shafts combined,

Then, $1030 \div 3 = 344$ Horse-Power per shaft or turbine,
and, $344 \times 33000 = 1032000$ foot-pounds of work per minute.
Therefore, $1032000 \div 778 = 1326$ units effective Heat Drop per minute.

The L.P. turbines are working in a vacuum of 21 in. at the initial end and 28 in. at the exhaust end. This heat drop is therefore given out by each L.P. turbine in working between pressures of $4\frac{1}{2}$ lbs. initial and $1\frac{1}{2}$ lb. final pressure (absolute), or within a total pressure drop of $4\frac{1}{2} - 1\frac{1}{2} = 3$ lbs.

NOTE.—21 in. $\div 2 = 10.5$ lbs., and 15 lbs. atmospheric pressure $- 10.5 = 4.5$ lbs. absolute.

Therefore, $3 \div 64$ Rows $= .046$ of a pound pressure drop per Row.

The effective heat value of low-pressure steam as utilised in turbines will perhaps be apparent from the foregoing figures, which are taken from actual practice.

To sum up—

In case (*A*) the H.P. turbine develops 900 horse-power working between pressures of 50 lbs. absolute and 9 lbs. absolute. So that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $50 - 9 = 41$ lbs.

Each L.P. turbine also develops 900 horse-power, working ***in a vacuum***, or between pressures of 9 lbs. absolute and 2 lbs. absolute, so that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $9 - 2 = 7$ lbs.

In case (*B*) the H.P. turbine develops 344 horse-power working between pressures of 25 lbs. absolute and $4\frac{1}{2}$ lbs. absolute, so that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $25 - 4.5 = 20.5$ lbs.

Each L.P. turbine also develops about 344 horse-power, working ***in a vacuum***, or between pressures of $4\frac{1}{2}$ lbs. absolute and $1\frac{1}{2}$ lb. absolute, so that the total pressure drop is 3 lbs., as $4.5 - 1.5 = 3$ lbs. The necessary heat drop is therefore obtained by this very small pressure drop, and the kinetic energy thus given up develops the required power.

SECTION II.

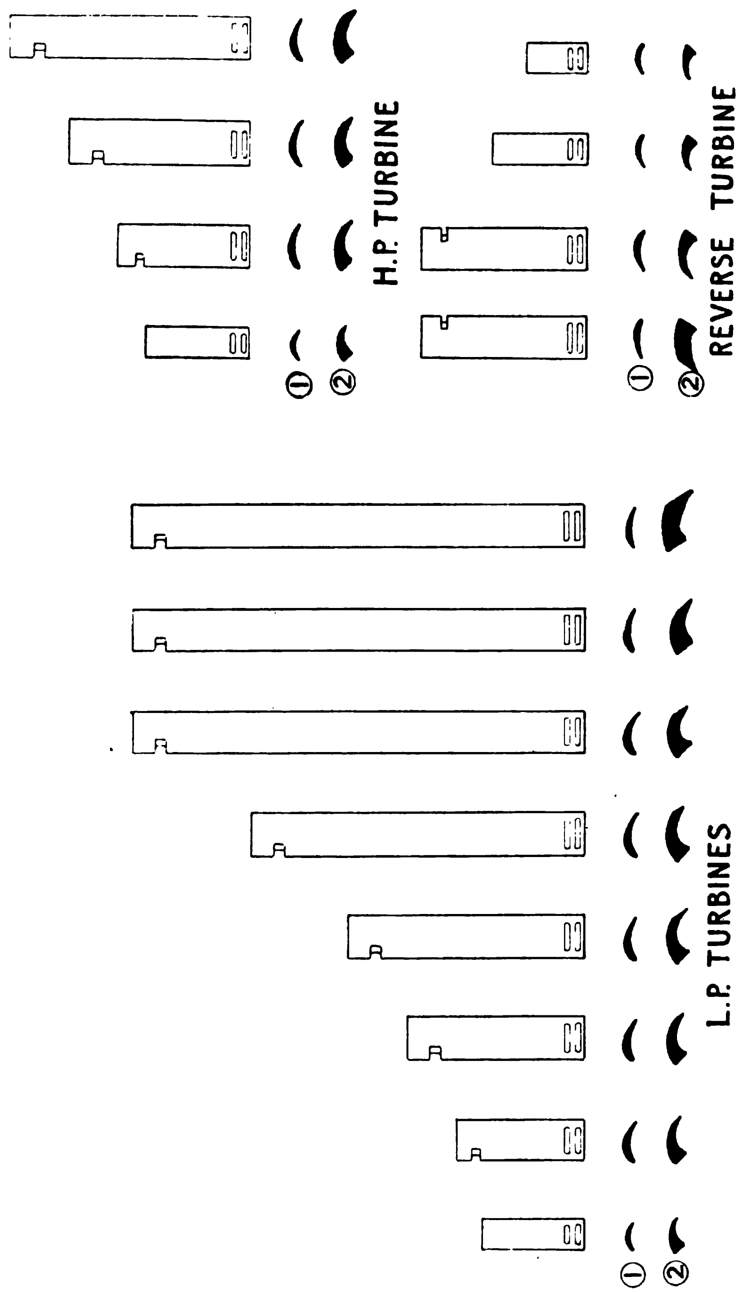
WORKSHOP PRACTICE.

THE turbine proper consists of two main parts; these are called the casing or cylinder, and the rotor.

The rotor is the revolving part, the casing or cylinder being stationary: the rotor consists of a round steel drum, into each end of which are inserted steel wheels or centres. Through the centre of these wheels spindles are fitted, which form the bearing shaft for the rotor. The outside of the drum is turned by machine, and at given distances grooves are cut, into which the brass blades are inserted. The casing is a cast-iron cylinder, which is turned inside, and is of increasing diameter at given distances in its length; grooves are also cut in the casing, and into these grooves brass blades of similar construction are inserted. The steam is admitted at the small end of the casing and passes through the blades. The reason for the increasing diameter of the casing is that the steam, after having done work, drops in temperature and pressure and increases in volume, so that to have a perfect arrangement of blades every row should be longer than the preceding one. In practice this has not been found practicable, and the blades are arranged in groups of different lengths and spacing, each group being termed an "expansion."

The blades do not lie parallel to the shaft axis or at right angles to it, but are placed at a slight angle, the shaft blades being at the opposite angle to those of the casing (see sketch). Each turbine *begins* with a ring of casing blades, and *ends* with a ring of rotor blades.

Usually in a high-pressure turbine there are 4 expansions, with 12 to 16 rows in each expansion, and in a low-pressure turbine there are usually 7 or 8 expansions of 6 to 10 rows in each. The blades in the rotor project outwards between the rows of blades in the casing, and the blades of the casing project inwards between the blades of the rotor. As explained elsewhere, the steam is admitted at the small end of the casing, and first passing through a row of casing blades is deflected so as to strike the rotor blades: after passing through the first row of moving blades it is again deflected, and strikes on the next row of fixed blades, which again deflects the steam so that it



Complete Set of Blades and Packing Pieces (half size).

1. Plan or edge view of Blades. 2. Plan of Packing Pieces.

NOTE.—The above are from the turbines of a steamer of about 5,000 I.H.P.

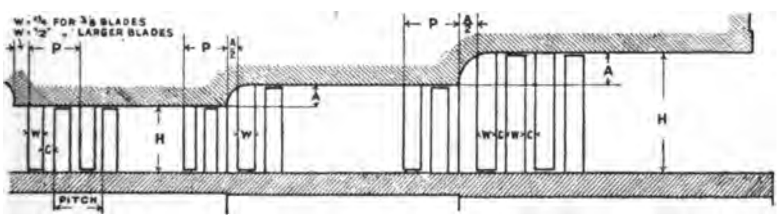
impinges on the next row of rotor blades, the steam reacting from one row to the other and flowing from the steam admission end of the turbine to the exhaust end. There are usually one high-pressure and two low-pressure turbines fitted. These turbines drive three shafts. The steam is admitted to the high-pressure turbine and exhausts from it into the two low-pressure turbines. Attached to the low-pressure turbines are the reverse turbines, which are used for going astern.

The economical running of turbines is greatly dependent upon the reduction of the clearance spaces to the utmost limit compatible with working conditions; this, of course, applies more or less to all steam engines, but in the case of turbines the steam blows continuously through from end to end over the tips of the blades, and this necessarily results in considerable loss of steam which, needless to say, is unavoidable.

TABLE III.—STANDARD BLADING DIMENSIONS.

Height (H)	1″	2″	3″	4″	6″	8″	10″	12″	15″	18″	21″	24″	30″
Width (W)	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	$\frac{5}{8}$″	$\frac{5}{8}$″	$\frac{3}{4}$″	$\frac{3}{4}$″	1″	1″	$1\frac{1}{8}$″	$1\frac{1}{4}$″
Pitch (P)	$1\frac{1}{8}$″	$1\frac{1}{8}$″	$1\frac{1}{4}$″	$1\frac{5}{8}$″	$1\frac{3}{4}$″	$2\frac{1}{8}$″	$2\frac{1}{4}$″	$2\frac{3}{8}$″	$2\frac{5}{8}$″	$3\frac{1}{8}$″	$3\frac{1}{4}$″	$3\frac{5}{8}$″	4″
Axial clearance (C) ...	$\frac{3}{16}$″	$\frac{3}{16}$″	$\frac{1}{4}$″	$\frac{5}{16}$″	$\frac{3}{8}$″	$\frac{7}{16}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	$\frac{9}{16}$″	$\frac{9}{16}$″	$\frac{5}{8}$″	$\frac{11}{16}$″	$\frac{3}{4}$″

NOTE.—While the above represents general practice, it is obvious that such a table is largely arbitrary.



Blade Dimensions

Construction of Rotors.

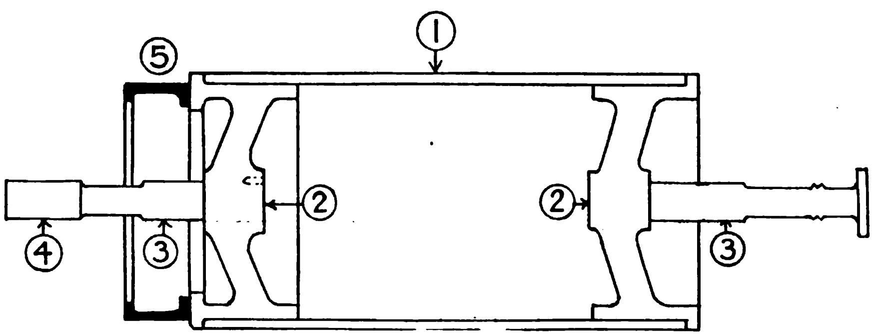
The rotor consists of—

1st, the drum.
2nd, the wheels.
3rd, the shafts or spindles.
4th, the dummy.

In large rotors, the drums being of great length are usually in two pieces, and have a centre wheel at the joint of the drum. The description which follows is of an H.P. rotor of the usual design of construction up to about 18,000 horse-power. The L.P. rotor will be described later.

The drums are worked out of the solid ingot, and are rolled out in large rolls. It may be mentioned that for the manufacture of

drums of large diameter there are only a few firms in the country who can undertake this work. The drums being delivered at the works rough turned, the next operation is the boring out of the inside to receive the wheels ; this is usually done either in a lathe when the drum is small, or in a large boring machine when the drum is of large dimensions. When binding the drum either in the lathe or machine care must be taken to have an equal pressure at the different points where the drum is held so as to ensure that no distortion takes place. To overcome this difficulty it is advisable to have two cast-iron half rings or glands lined with wood ; the wood is bored out to the size of the outside of drum, and then the drum is put inside bottom half of ring, top half put on, and screwed up.



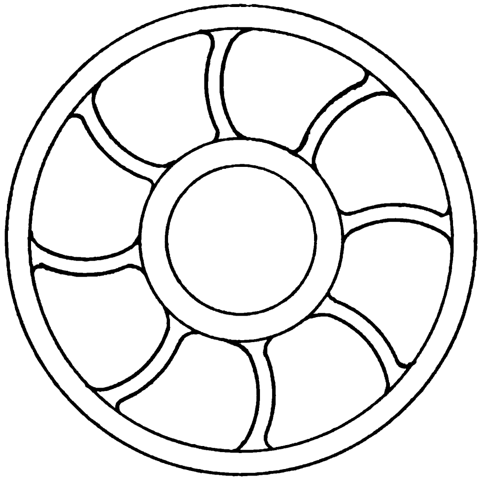
H.P. Rotor Drum.

1. Drum. 2. Wheels. 3. Spindles. 4. Thrust. 5. Dummy.

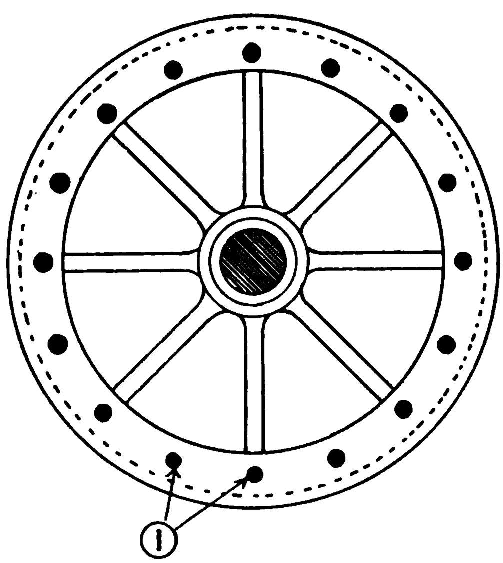
The drum being bored, the wheels are now brought forward. The wheels are of steel, either cast or forged, and there are several different designs, usually the cast-steel wheels have a number of ribs, and these occasionally are of unequal number so as not to throw a transverse strain on the wheel when cooling out after casting. Another type of wheel is of the forged steel type, of disc form, forged from the ingot and turned all over, this ensuring a perfect balance. It may be mentioned that this type of wheel was adopted in the "Mauretania." In large wheels of the cast type the outer rim is sometimes cast with three spaces, or joints in the outer rim, so as to avoid contraction cracks. These joints are afterwards filled up with steel liners bolted to a flange on the wheels.

The wheels are bored out in the centre or nave to receive the shafts or spindles, which are turned on the ends, to sizes suitable for shrinking.

After wheels and shafts are machined the wheels are put into a drying stove in the foundry, if there is no suitable stove in the erection shop for this work, as it is advisable that the wheels should be equally heated up all over. The wheel is then heated up and expanded



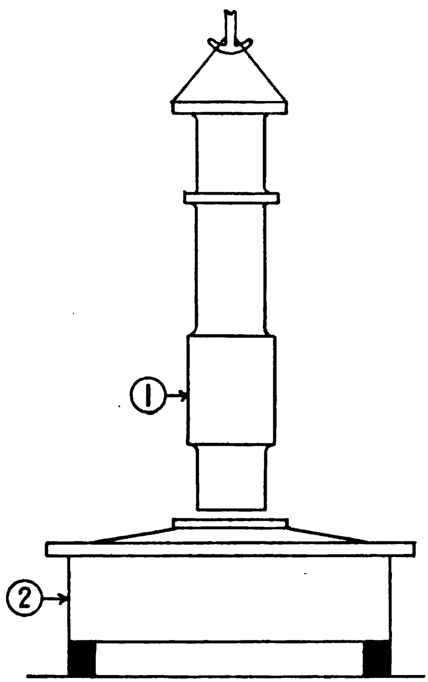
"Wheel" or "Centre."
With Curved Arms to allow of Expansion when Heated.



"Wheel" or "Centre."
1. Bolt Holes for Dummy.

to $\frac{1}{32}$ in. larger than the diameter of the shaft which is to go into it. The spindle must be slung dead plumb, and care should be taken to have the wheel also truly levelled up.

After taking wheel out of stove or furnace the spindle is suspended above and gently lowered until it enters wheel, then lowered as smartly as possible until spindle rests on shoulder. In cooling out the wheel it is preferable to use jets of compressed air in preference to water, and to cool down part of wheel nearest shoulder on spindle first so as to ensure the shoulder being close to wheel, as in contracting while cooling this may sometimes be open. Compressed air for cooling is preferable to water for the reason that it does not run down the wheel, and only cools out the part intended. The wheel being



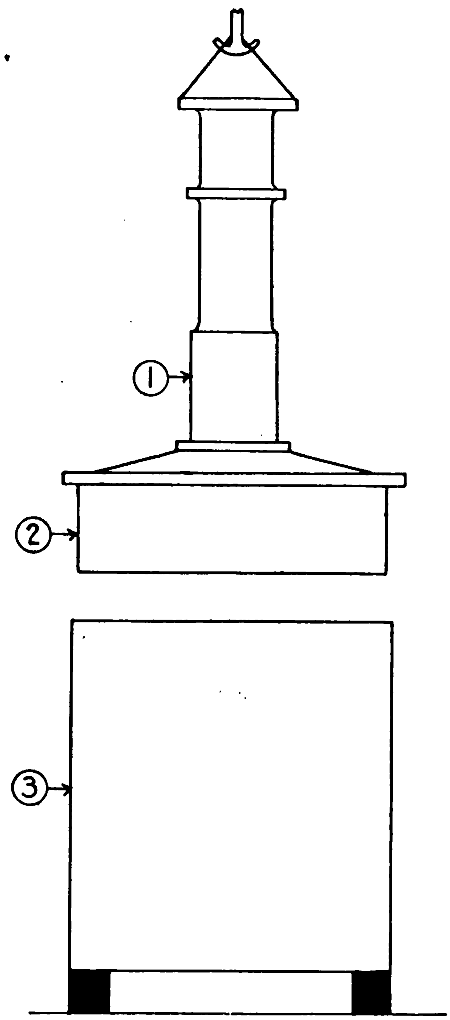
Method of Shrinking on Spindles to Wheels.

1. Spindle. 2. Wheel.

cooled is now put into machine and holes bored half in shaft and half in wheel. Pins are screwed into these holes, the heads of pins being riveted over to prevent slackening back.

In large rotors, instead of screwed pins, round driving keys are fitted, and the end of the spindle is screwed and a round nut put on which covers keys. The nut is prevented from slackening back by stop pins being fitted half into shaft and half into nut. The wheels are now put into lathe again and outside diameters turned to suit inside bore of rotor drum, and the same amount as before allowed for shrinkage. The dummy is a steel ring which is attached to the steam inlet end of

the rotor. There are two kinds of dummies, facial and radial. Facial dummies are usually adopted on the low-pressure and high-pressure



Method of Shrinking on Wheel to Drum.

1. Spindle. 2. Wheel. 3. Drum.

NOTE.—The wheel is shown suspended in readiness for lowering into the drum which has been heated and expanded to receive it.

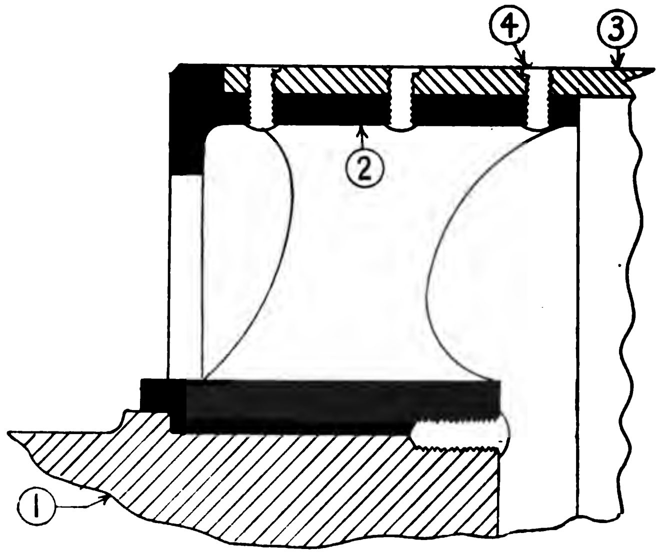
ahead turbines, the radial dummy is usually fitted on the astern turbine.

The use of the dummy is to prevent the steam leaking to the

glands at the end of the casing, and also to prevent the steam passing through the interior of the rotor to the exhaust, instead of doing legitimate work by passing through the turbine blades.

Shrinking or Building of Rotors.

The rotor drums are erected, if the rotors are small, on suitable blocks; if large, then it is necessary to have a shrinking pit so that the spindle of rotors will clear when turned up on end. The drums are set up and plumbed and then heated with either blow lamps in the case of small rotors, or by gas rings when constructing large rotors,



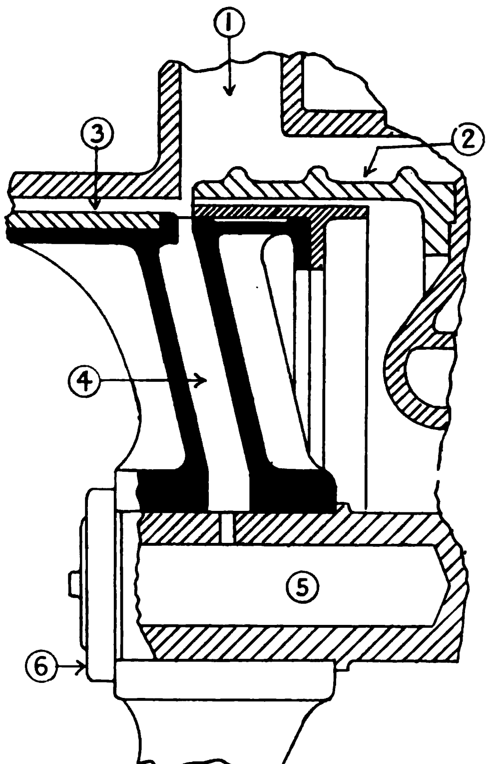
Method of Connecting Wheel, Spindle, and Drum.

1. Spindle. 2. Wheel. 3. Rotor Drum. 4. Screwed and Riveted Pins, with small Feather to Prevent Slackening Back.

the drum being heated and expanded so that a gauge $\frac{1}{32}$ in. larger than the wheel will pass clear down to a little further than the depth of the wheel. The wheel and spindle are slung, care being taken to ensure them being plumb, and the wheel is entered into drum and lowered into place.

The drum is now cooled out at the top first in the same manner as before by compressed air. The drum is now marked off for the rivets or pegs which bind it to the wheel; the rivets are pitched zig-zag, a few holes are bored, and temporary bolts put into same, after which the drum is turned upside down with the rotor shaft either in the pit or, if a small rotor is being constructed, between wood blocks. The other wheel is now shrunk in like manner, bolts or pins put in

and rotor canted down. The rotor now goes to boring machine for the cutting of the rivet holes, after which rivets are put in and closed up. The rotor is now ready for turning. After rotor is put in lathe care should be taken to see that it is true, and if not the centre should be adjusted to make it so. It is then turned all over the body to the required given diameter and marked off for grooving. This operation



Hollow Cast Turbine Wheel (Naval type).

1. Steam Inlet. 2. Dummy. 3. Rotor Drum. 4. Hole to allow Admission of Steam to Hollow Spindle for Uniform Heating up and Expansion. 5. Hollow Spindle. 6. Cap.

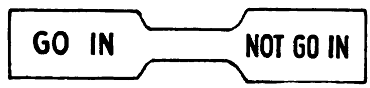
NOTE.—The above arrangements have been devised to obviate the risks of unequal expansion when heating up.

is done in different works in different ways, one method being to have the casing or rotor grooved first, the rotor put into place and grooves marked off relative to the grooves in part which has already been grooved.

Another method is the use of length sticks marked off from the drawing, the position of grooves, &c., relative to the shoulder on bearing and position of steam glands.

A similar length stick is also prepared for the casing grooves, and by applying the two length sticks together the position of rotor and casing blades can be ascertained with accuracy.

Turning of Rotors.—The rotor being marked off, gauges should be prepared for the width of groove and depth of groove. It is preferable to have two gauges for the width of groove, one to "go in" and one "not to go in." The "not go in" gauge should be .003 to



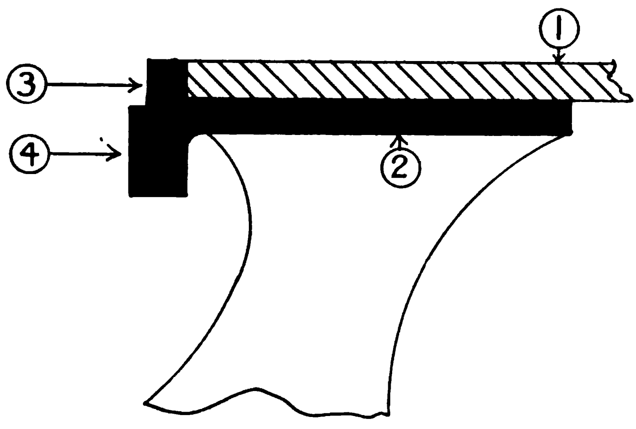
Test Gauge for Width of Grooves.

.004 larger than "go in" gauge. This ensures the groove being of proper width to suit the blading. A complete set of gauges made for blading are supplied (standardised), the terms of which are as follows:—

BLADING.

130 B, 140 B, 240 B, 250 B, 251 B, 260 B, 261 B, 271 B, 272 B, 280 B, 281 B.

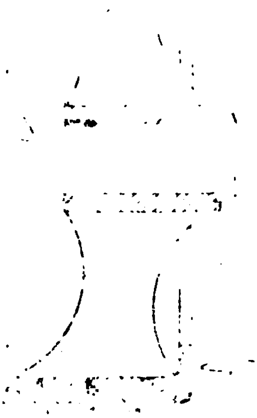
Packing pieces are of similar number, and have the describing



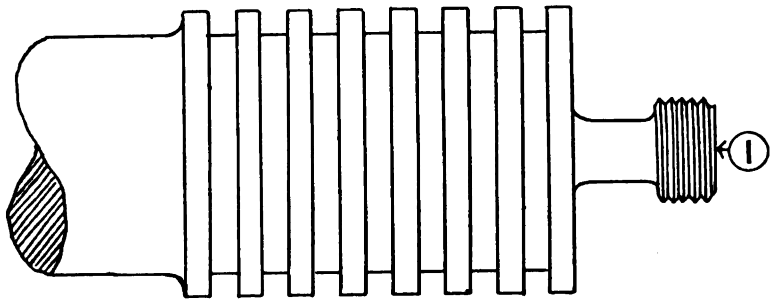
Wheel and Drum.

1. Rotor Drum. 2. Wheel. 3. Recess for Dummy Spigot. 4. Face for Dummy.

letter C and S after the number instead of B. B represents the blades, C the packing pieces for cylinder or casing, and S the packing pieces for rotor. The grooves are turned, and on each side of same serrations are made, an indentation of V shape on each side of the groove. In shallow grooves the serrations are two in number, and in deep grooves four serrations are formed on each side of groove. These serrations are to assist in holding the blading in place, and will be further described under the heading of "Blading." The serrating tool

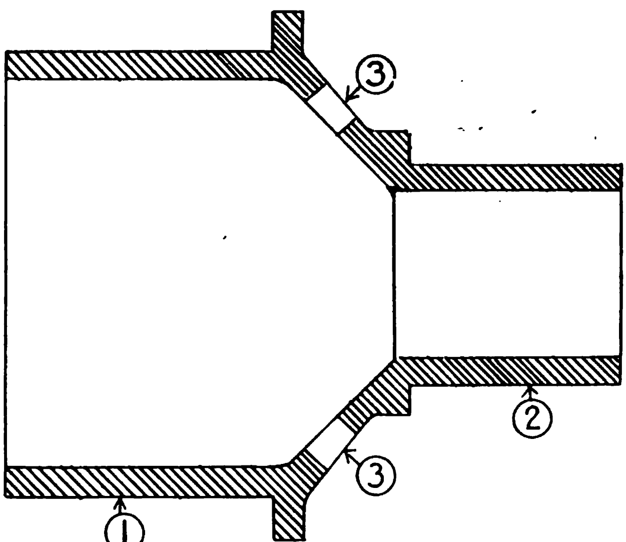


is usually made so as to have a cutting edge on each side; one side is cut, and then the tool is turned against the opposite side of groove and serrations formed. After grooving, the forward end of the rotor,



Thrust and Counter Gear (on Forward End of Spindles).
1. Worm connecting to Counter Gear.

that is the face of the forward wheel, is turned up to receive the dummy, which is bolted to this face. This face is spigoted to fit the dummy recess, thus ensuring the one being central to the other. In

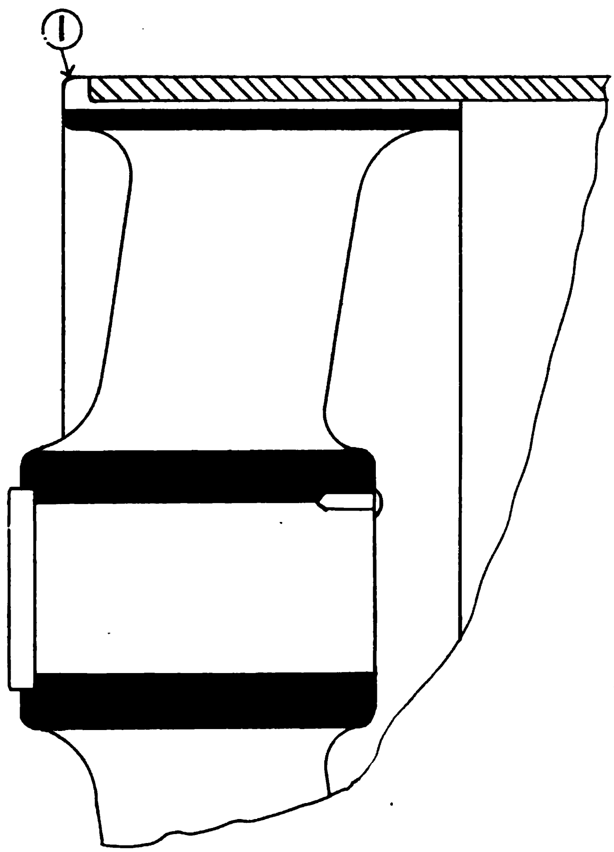


Junction Wheel (for connecting up the L.P. Ahead and Reverse Rotor Drums).
1. L.P. End of Wheel. 2. Reverse End of Wheel. 3. Holes in Wheel.

some cases the dummy is part of the wheel, but it is preferable to have a separate part, as it may be required to renew same, owing to pitting taking place and destroying the grooves in the dummy.

The bearing spindles are now turned up and finished. The part

of the spindle nearest to the wheel is of larger diameter than that forming the bearings, and this part is grooved for the steam gland packing. These grooves are usually $\frac{3}{16}$ in. deep × $\frac{1}{8}$ in. broad, and into these brass strips are driven, thus forming what is termed "labyrinth" packing. This is described more fully under the heading of "Steam Glands." At the end of the part which forms the bearings

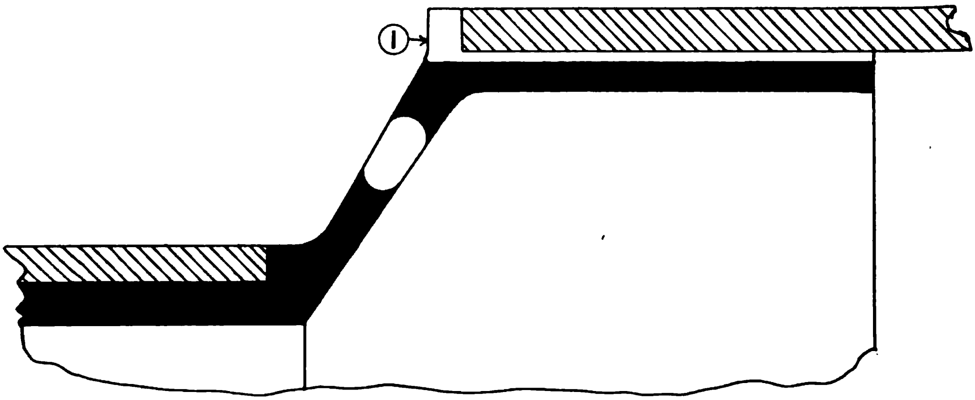


Wheel.

1. Drain from Inside of Drum.

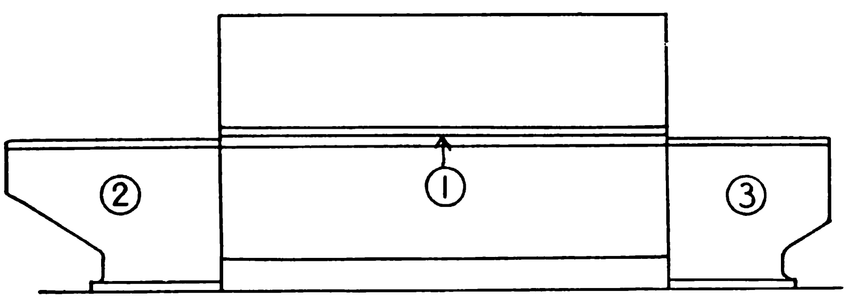
a series of V indentations are made; these act as oil deflectors, and prevent the supply of oil which is forced through the bearings from escaping along the shaft. A low-pressure rotor differs only from a high-pressure rotor in that in average size turbines the rotor consists of one large drum and one small drum. The small one is for reversing purposes, and is usually at the after end of the low-pressure rotor. The rotor is built in the same way as an H.P. rotor, and has a con-

necting wheel between the large drum and the small one; this wheel is termed a "junction wheel." The wheel is turned all over, and is of two diameters, one to go into the aft end of the L.P. drum and the other to go into the forward end of astern or reverse drum. Through the wheel holes are bored, which allow any steam which may get into



Junction Wheel (for connecting up the L.P. Ahead and Astern Drums).
1. Drain from Inside of Drum.

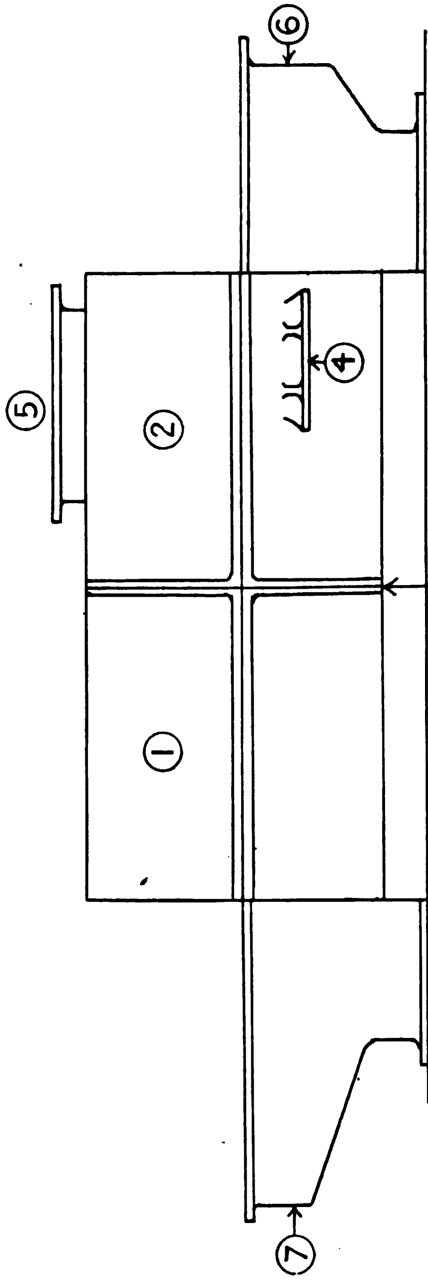
the inside of the rotor to escape to the exhaust. It will be seen that the interior of an L.P. rotor is thus in a vacuum. The rotor drum is pinned to the wheels in a similar manner to the H.P., and is turned up and grooved. The rotors are parallel, and the casing is stepped to suit the different lengths of blades of each expansion.



H.P. Turbine Casing.
1. Horizontal Joint. 2. Bearing Stool and Thrust. 3. Bearing Stool.

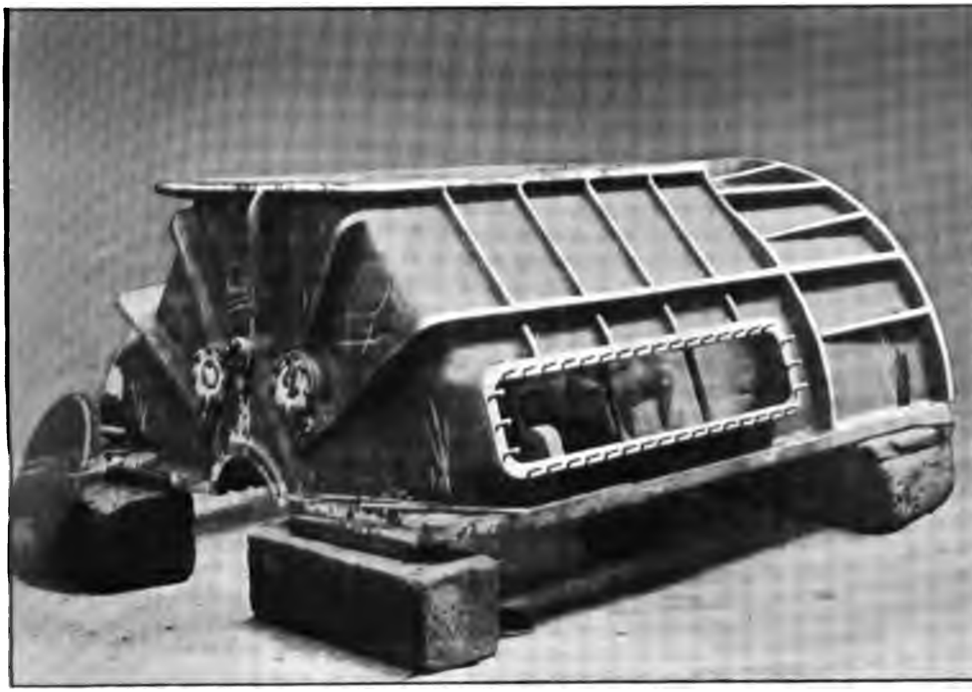
Turbine Casings.

Turbine casings or cylinders are of cast iron. There are several different designs in regard to the ribbing and strength of same. In merchant steamer work they usually have heavy bulb ribs running longitudinally and circumferentially about the shell. The high-

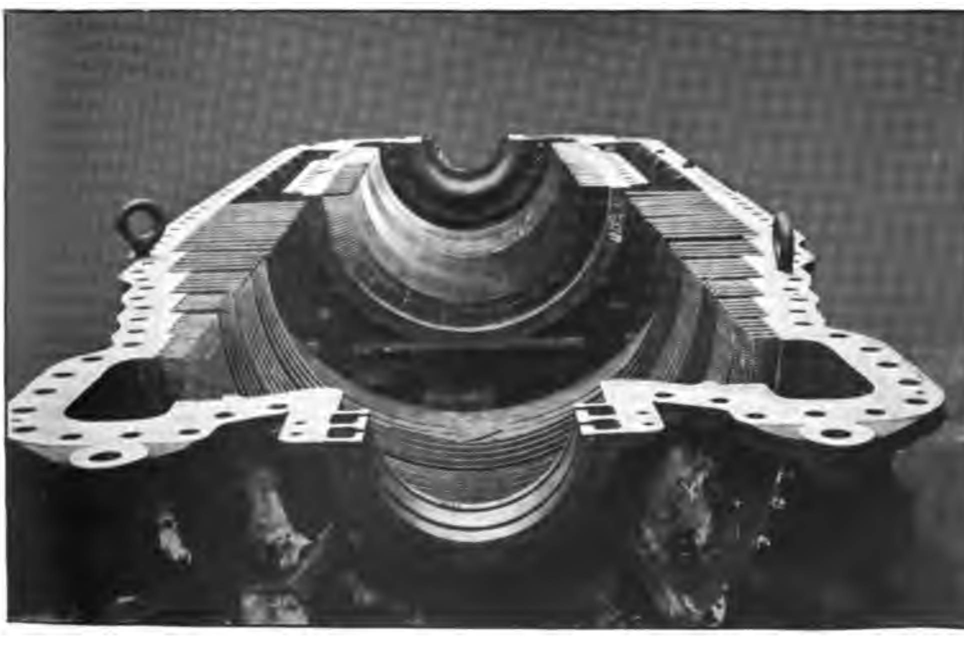


L.P. Ahead and Reverse Turbine Casing.

1. Forward top half section.
2. After top half section.
3. Vertical Joint.
4. Supporting Feet.
5. Exhaust to Condenser.
6. Bearing Stool.
7. Bearing Stool and Thrust.



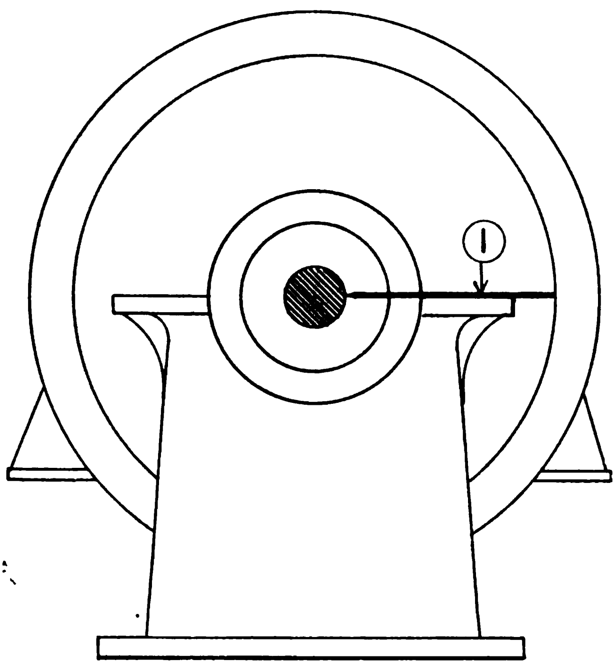
External View of Upper Half Casing of L. P. Ahead and Reverse Turbines.
(33 Knot Destroyer)
The Opening shown is that of the "Access Door" to Astern Casing.



Upper Half Casing of L. P. Ahead and Reverse Turbines.
(33 Knot Destroyer).

[*To face page* 76.

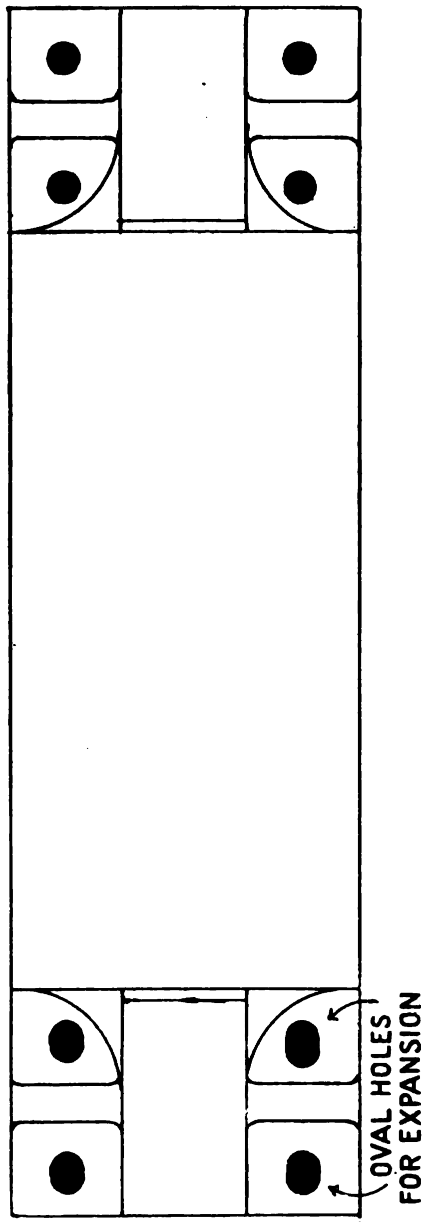
pressure casing in average size turbines usually consists of two parts, the top and bottom halves of casing. The low-pressure casing is usually in four pieces, the top half being made up of two parts, and the bottom half of two parts. These two parts are bolted together, forming a circumferential joint. In large turbines the casing sometimes consists of six or eight pieces—this is to ensure good castings. When casings are delivered from foundry to shop, they should be calipered all over to ascertain thickness of metal, and then put on surface table and drawn in. The horizontal joint is then planed in both halves, and marked off for boring bolt holes. In an L.P.



Gauging Bore of Casing.

1. Gauge.

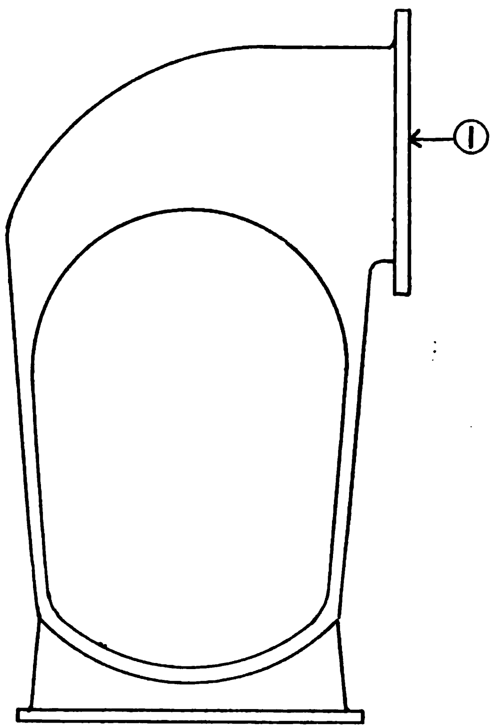
casing, there being four parts, after the horizontal joint is machined and bored in one section, and four corner holes in the same are widened and temporary bolts fitted; the casing is then sent to the turbine-boring machine, and inside of casing rough bored to $\frac{1}{8}$ in. smaller than the finished size. The same operation takes place with the other section, after which the bolt holes in circumferential joint are bored. The high-pressure casing goes through the same operation of rough boring, and is now ready for water testing. This casing is usually tested to from 230 to 250 lbs. per sq. in. The forward end of the low-pressure casing is tested to 50 lbs. per sq. in., and the aft end to 30 lbs. per sq. in.



Plan of Casing showing Oval Bolt Holes at Forward End for Expansion.

The after end is bolted rigidly to the seating of the steamer.

After the casings are water tested, they are "steamed" for twelve to twenty-four hours at a pressure of from 20 to 50 lbs. This is of great importance, as it tends to thoroughly expand the metal, and if not properly carried out, may result in the distortion of the casing when running the turbines, and cause destruction of blades. Each section of the low-pressure casing being separately tested, the two halves are bolted together, and steamed in a similar manner. After steaming, the joints of the casings are bedded, and in the case of the low-pressure casing, the circumferential joint is made. This

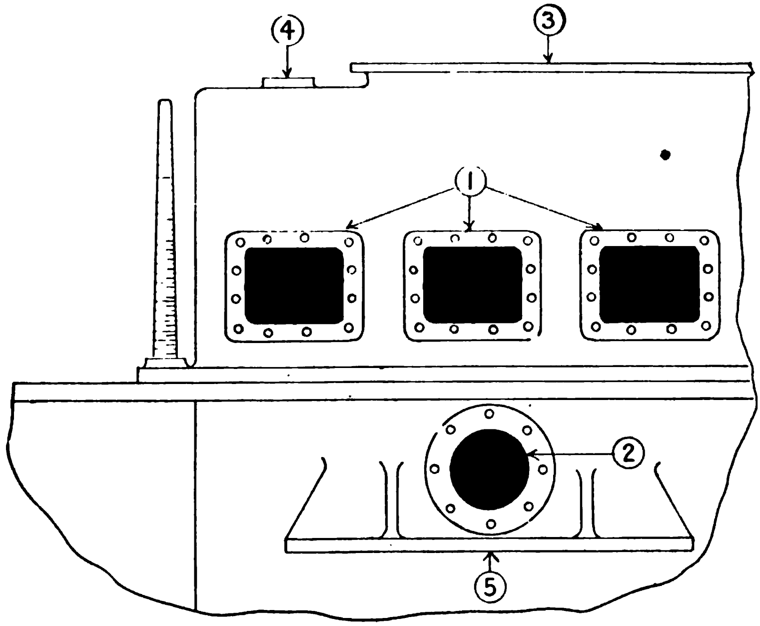


Turbine Condenser.

1. Exhaust from L.P. Turbine.

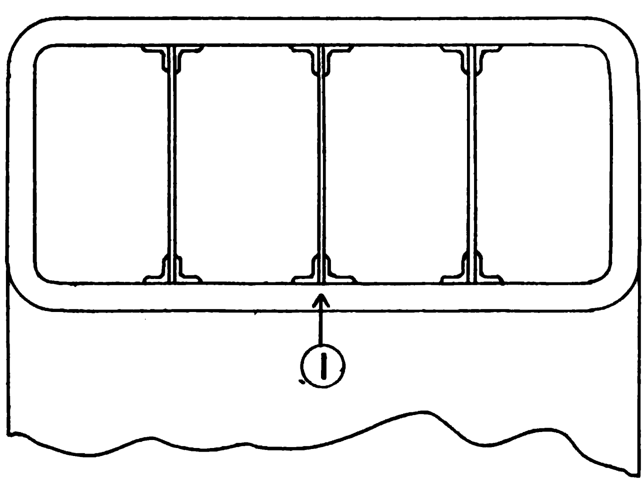
is usually made of mastic cement or red lead put on in the consistency of thick paint. The casings are now ready for final boring and grooving. In setting up in the machine, care should be exercised to ensure casing joint being exactly central to boring bar.

The casings are bored out to gauges applied from boring bar, and are marked off for grooving. The ends of the casing where the dummy is bolted to, and also the face where the astern cylinder is bolted to, in low-pressure turbines, are turned up. Usually in average size turbines the main bearing stools are part of the bottom



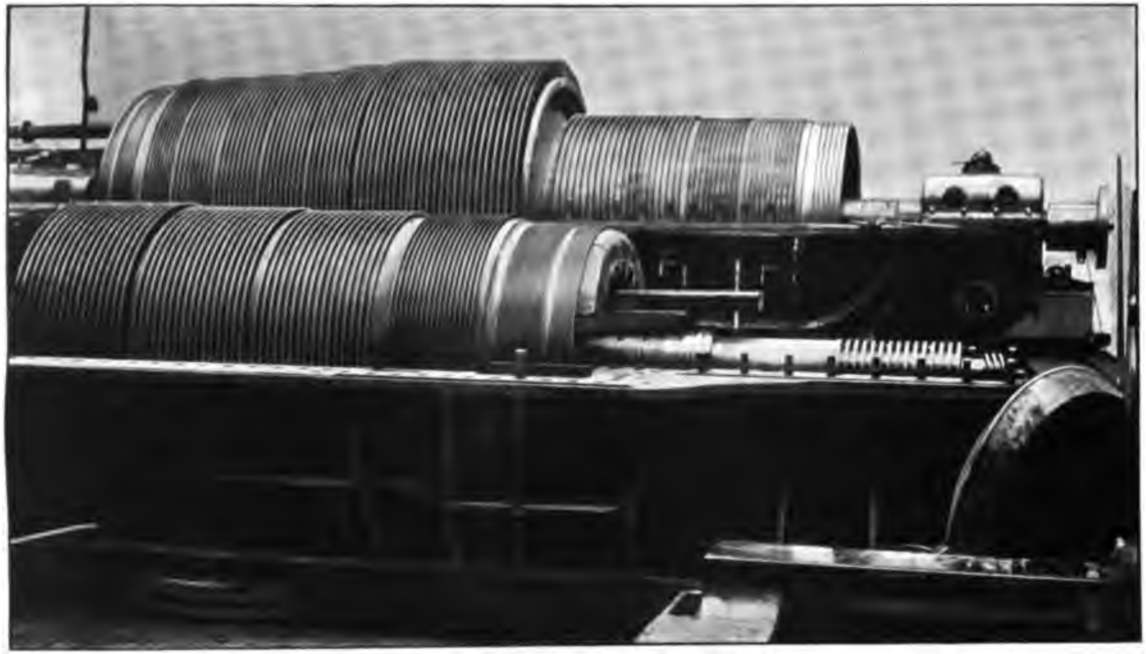
After End of L.P. Casing.

1. Access Doors to astern casing.
2. Access Door to bolts of astern cylinder feet.
3. Exhaust to Condenser.
4. Relief Valve.



Exhaust Casing.

. Plate and Angle Iron Stays.



Views of H.P. and L.P. Turbines.

Showing Ahead H.P. Dummy, and the four H.P. Expansions, also the L.P. Ahead and Reverse Dummies, and the eight L.P. Ahead Expansions, and four Reverse Expansions.

NOTE.—Each H.P. Expansion contains sixteen rows of blades, and each L.P. Expansion contains eight rows of blades.

[*To face page* 80.

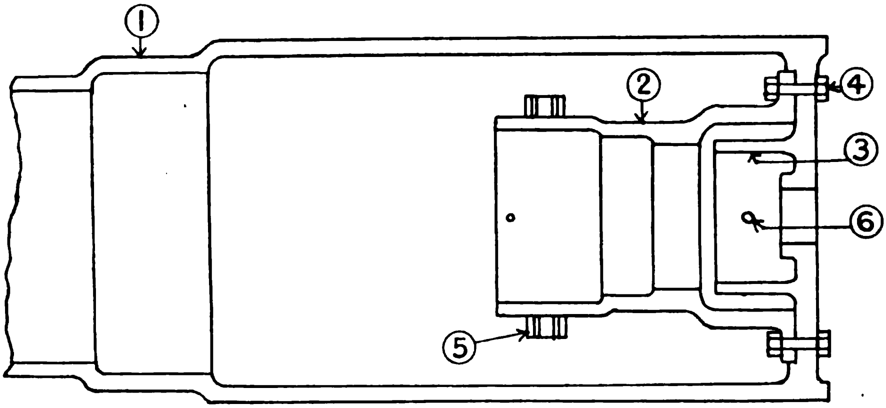
View of L.P. Turbine and Reverse Turbine in Lower Half Casing.

Showing (left) "Wheel" or "Centre" in Lower Half Casing, Ahead Dummy Piston, Ahead Expansions (eight in number), Reverse Expansions (four in number), and Reverse Dummy Piston.

NOTE.—The last three Ahead Expansions have blades of equal height, but of wider spacing, and the last two Reverse Expansions have blades of equal height, but of wider spacing. The steam pipe shown is only that fitted in the shop for "steaming" purposes.

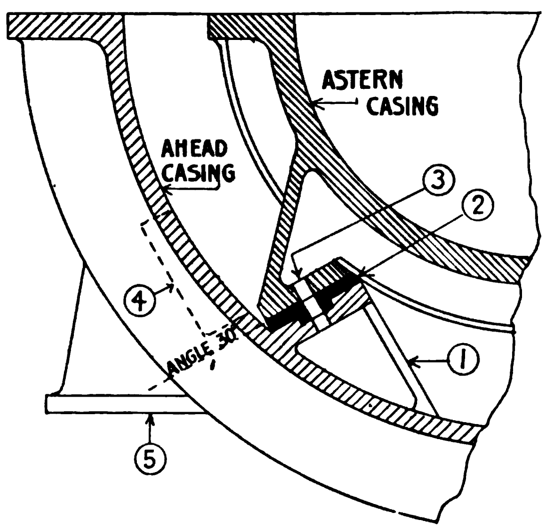
[*To face page* 80.

half of casing, and in this case they are bored out in this machine, thus ensuring the bearings being concentric to the bore of the casing.



L.P. Casing and Reverse Casing.

1. L.P. Ahead Casing. 2. Reverse Casing. 3. Reverse Dummy. 4. Bolts of Reverse Casing. 5. Feet of Reverse Casing. 6. Dummy Casing Drain.



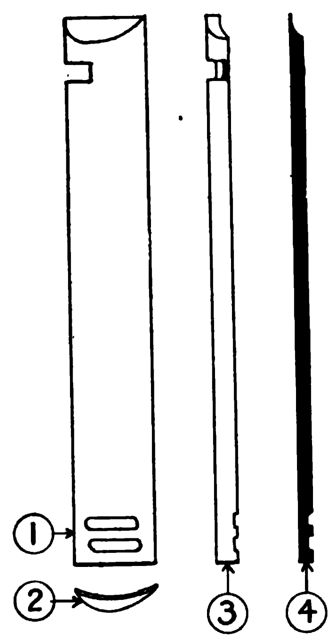
Astern Turbine Casing.

1. Bracket cast on L. P. Ahead Casing to receive reverse casing feet. 2. Brass Chock. 3. Hole ⅛ in. larger than Bolt to allow of expansion. 4. Access Door. 5. Feet of L. P. casing.

For turbines up to 18,000 horse-power the usual arrangement is to have the astern casing or cylinder inside the after end of the low-pressure. The astern casing is bolted to a flange inside the low-

pressure casing, and is also supported by diagonal feet on each side which sit on a ledge on the wall of the low-pressure casing. A brass chock is fitted under this foot, and a bolt put in, a loose fit, so as to allow the astern cylinder to remain central when expansion comes on after casing is heated up. The astern cylinder is turned and grooved in a similar manner, and is water tested to a pressure of from 100 to 150 lbs. per sq. in.

Blading.—The blading for Parsons' turbine is of brass, and is manufactured by various firms: it is usually delivered in lengths of from 5 to 6 ft. The turbine proper being formed of blades of

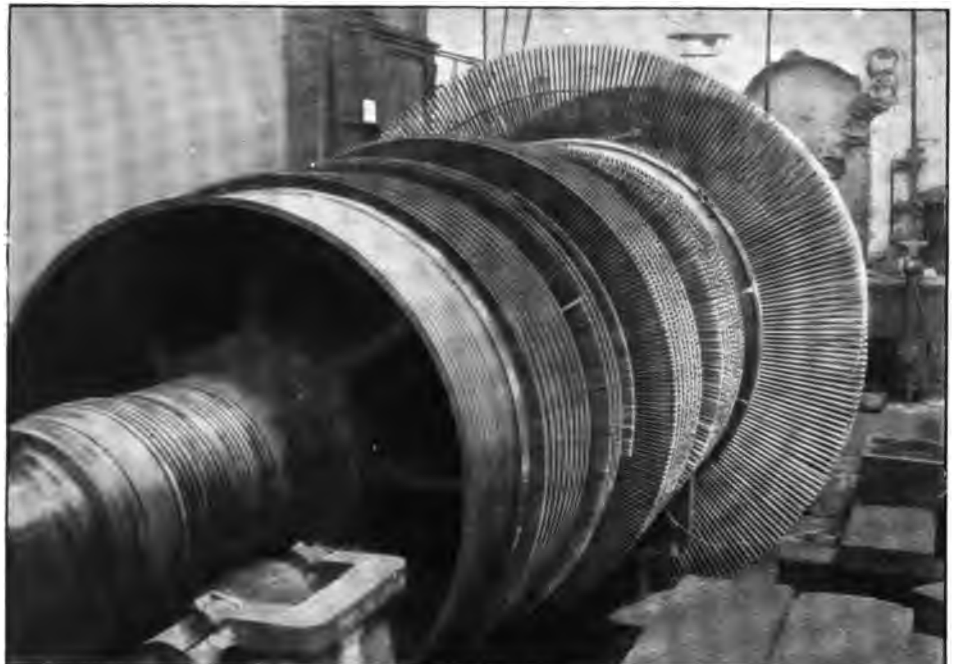


Blade with Thinned Tip.

1. Elevation of Blade.
2. Plan of Blade
3. Edge view of Blade.
4. Section of Blade.

various lengths and spacing, which are termed "expansions," the blades are cut to lengths in a machine which shears them off and at the same time stamps a double or treble groove on the end.

These serrations are to assist in holding the blade in place (see sketches). The blades being of various lengths, the rotor is left parallel and the casing stepped to suit this. After the blades are cut to length they pass to a machine which cuts a groove about ¼ in. from the end of blade remote from the serration, and into this groove the binding wire is fitted. From this machine the blades pass to a machine which thins down the point on tip of blade to about .001 in.



Rotor Partly Bladed.

(33 Knot Destroyer).

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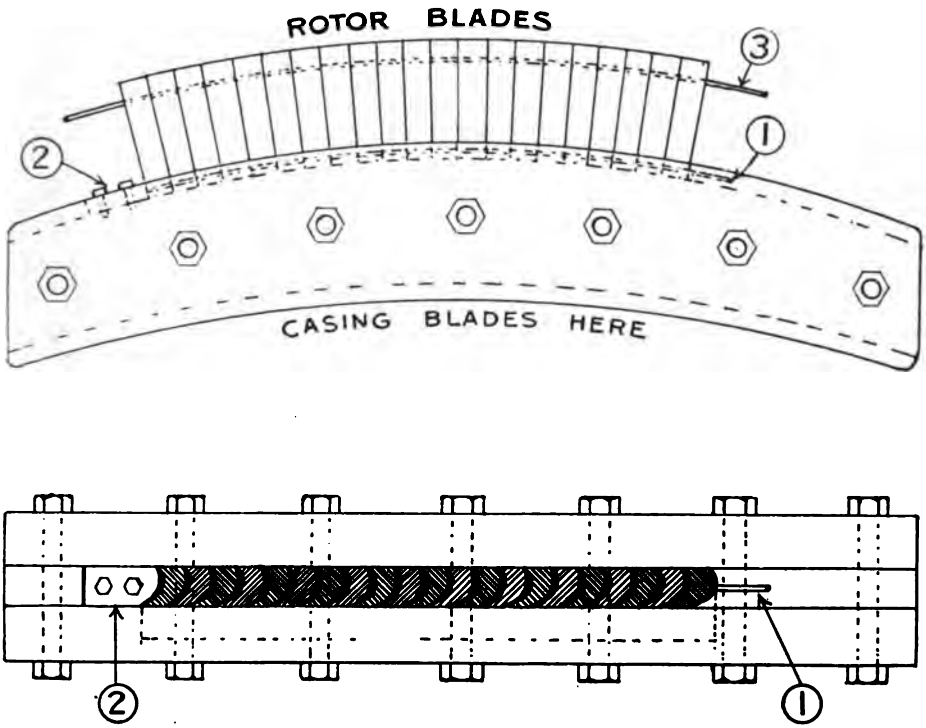
so that in the event of blades coming in contact with the casing or rotor they will simply turn over or wear down without serious damage resulting. There are several different systems of blading, the original system being to blade the rotors and casings by hand, but this has been superseded to some extent by what is termed "segment" type blading. There are two types of segment blading, "Parsons'" and "Willans & Robinson's." The Parsons segment blading is built up in

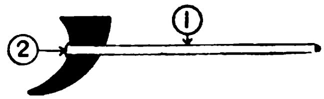


"Segment" Block for Blading.

1. Brass Wire through Blades and Packing Pieces. 2. Stop Piece pinned down in Groove. 3. Brass Binding Wire lashed and soldered to Blades.

two cast-iron plates which are bolted together, and have a groove, equivalent to the groove in casing or rotor for each expansion, turned in them. Usually one side of the block is for rotor and the other side for casing, the block being cast concave on one side and convex on the other. The blades have an $\frac{1}{8}$-in. hole drilled through them at the root, through which an $\frac{1}{8}$-in. brass wire passes. Between each blade there is a packing piece section. The packing piece is of soft brass and is delivered in lengths, which are cut up in revolving circular saws, in sections equal to the depth of the groove into which they are to be fitted. It is preferable to leave packing pieces

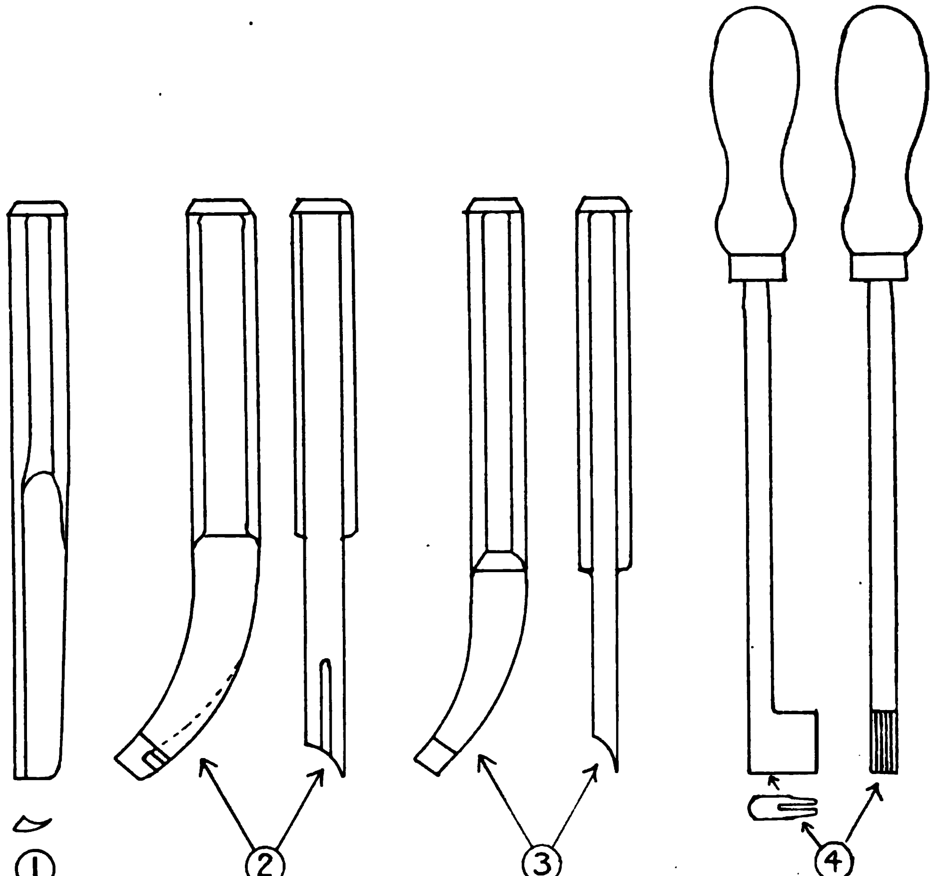
$\frac{1}{32}$ in. deeper than groove, so that when caulked into place they will then be flush. Through these packing pieces an $\frac{1}{8}$-in. hole is also drilled at



Segment Blading

(First Operation.)

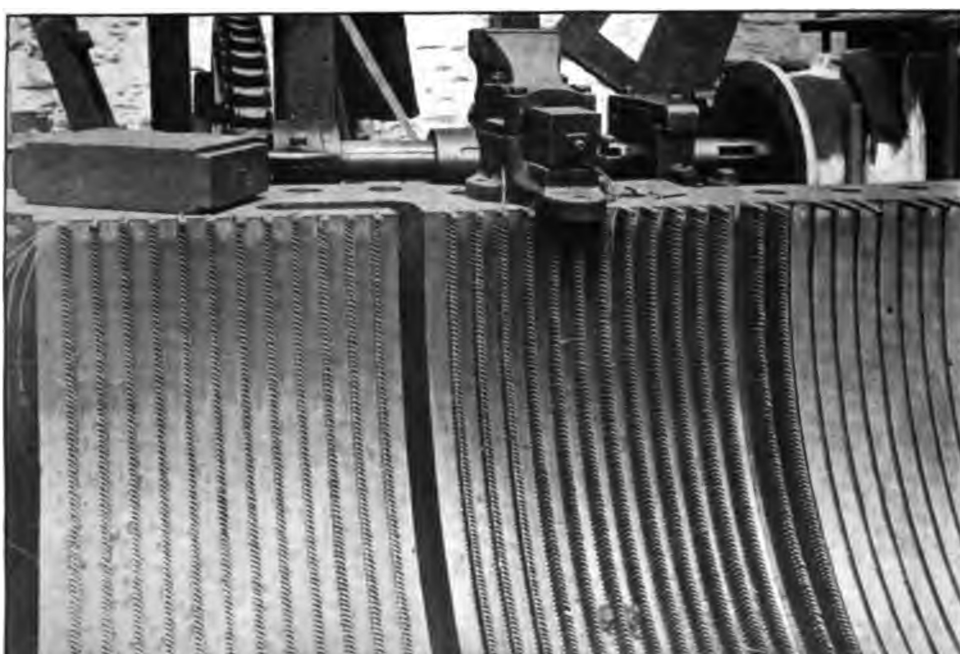
1. Brass Wire $\frac{1}{8}$ in. diameter passed through Blades and Packing Pieces.
2. Brass Wire riveted over Packing Piece.



1. Caulking Tool for Blades. 2. Driving-up Tool for "Segment" Blading.
3. Driving-up Tool for Standard Blading.
4. Tool for "Setting" Blades (sometimes called a "Twister").

No. 1 is employed in caulking up the packing pieces and blades. Nos. 2 and 3 are employed in driving up the blades and packing pieces endways previous to caulking. No. 4 is employed in twisting or setting the blades to the correct angle and position after caulking.

L.P. Lower Half Casing and Reverse Casing.



Inside View of Lower Half Casing.
The Blade "Stoppers" are seen at the top of each row.

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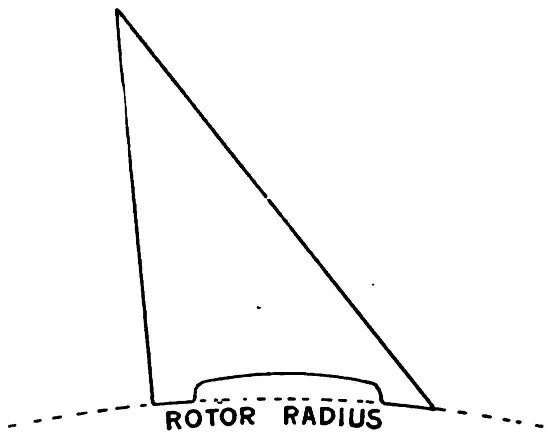
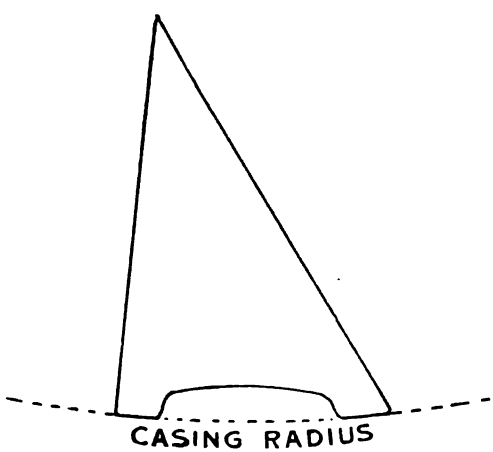
the requisite angle in a machine, in which are two dies of the same shape as the back and front of the packing piece. These catch the packing piece and hold it at the proper angle, so that the hole is in



Set-Square for Setting Rotor Blades.

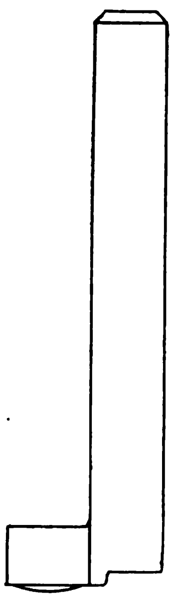
line with the hole in the blade. After being bored the packing pieces are annealed by heating to a dull red heat and allowed to cool out. A packing piece is taken, and through it a length of $\frac{1}{8}$-in. brass wire, a



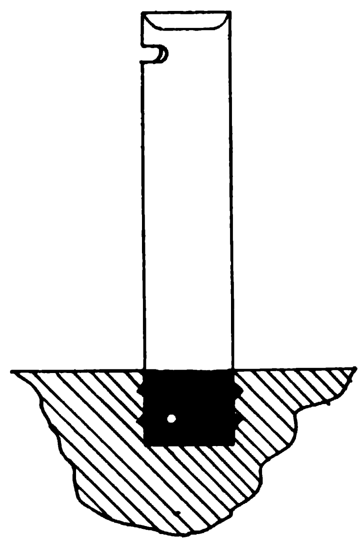
Set-Square for Setting Casing Blades.

little longer than the segment required, is passed and the end of the wire riveted over ; this is put into the groove in the building block and driven up against a stopper pinned into grooves. A blade is then

passed on to wire, next a packing piece, then a blade, and so on. About four blades and five packing pieces are put on, and the lot driven up against the stopper. This is done by a caulking tool having a groove cut on its under side so as to clear the wire which passes through the blades and packing pieces (see sketch). This operation is carried on, care being taken to see that all blades and packing pieces are resting on bottom of groove in building block until the required length of segment is built. At the end of segment two or three packing pieces are put in and driven hard up against last blade so as to ensure segment remaining firm until it is completed. A segment starts with a packing piece and ends with a blade. The



Caulking Tool for "Segment" Blading.

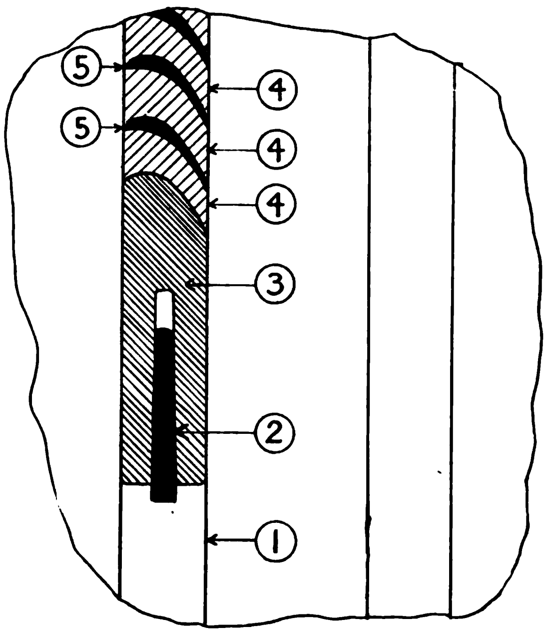


"Segment" Blading.

View of Blade caulked into groove with ⅛-in. brass wire passing through Blades and Packing Pieces.

blades are now trued up or spaced. This is done by using a set-square suitable to the radius of the casing or rotor for which the segment is intended. After trueing up the blades the binding wire is fitted into groove at top of blades, and in blades of over 3 in. in height this wire is usually laced with copper wire. The wire is now soldered with silver solder, using a gas blow lamp and a flux of fine powdered borax. In segments of long blades there is usually a middle binding wire which is soldered into place, but is not laced. After the segment is completed the packing pieces at the end are taken out, and the wire is cut and riveted over the hole at the root of the blade. One segment for each row is usually left with two or three blades unsoldered so that segment may be made the proper length for closing row. The

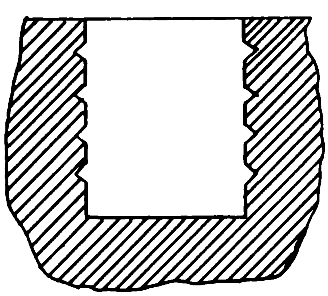
original Parsons' method of blading in the case of a rotor is as follows. Into the groove a stop piece is put, this stop piece being held in place



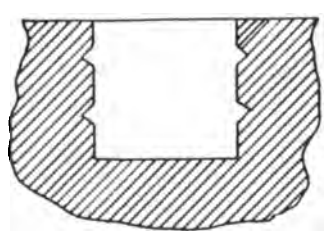
Method of Blading Rotors.

1. Groove in Rotor Drum.
2. Steel Wedge to Lock "Stop."
3. "Stop," against which the first Packing Piece is Butted.
4. Packing Pieces.
5. Blades.

by having a saw-cut in one end of it into which a steel wedge is driven, thus binding it against the sides of the groove. The blades and pack-



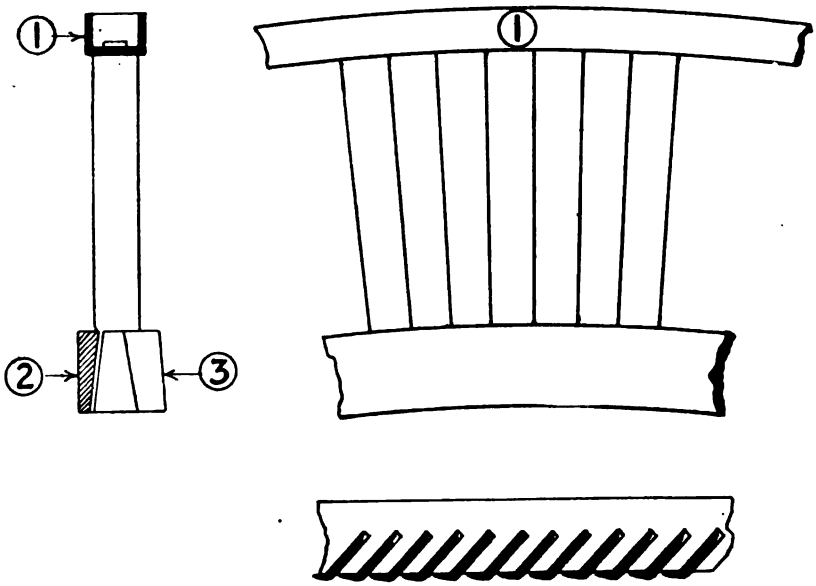
Groove and Serrations for Extra Long L.P. Blades.



Groove and Serrations for Short and Medium Blades.

ing pieces are driven up against this, four or so blades at a time, until the row is completed. It is then tuned and soldered in a similar

manner as in segment building. The Willans & Robinson segment building consists of a base piece or foundation ring, having saw-cuts on an angle equal to the angle of the blade of which segment was to be built. The blades are cut to length in the usual way, and then stamped on the bottom to the shape of an L. The straight side of the blade is put into the base ring with the tail lying against the side of ring, and the whole put into a building block, which is screwed up against blades and rings, thus binding same. The blades are tuned and soldered in a similar manner as in Parsons' blading. Over the tips of the blades a "shroud" is fitted of channel shape ⊔. This shroud is of brass about $\frac{10}{1000}$ thick, and in the event of blades coming



Willans & Robinson System of Blading.

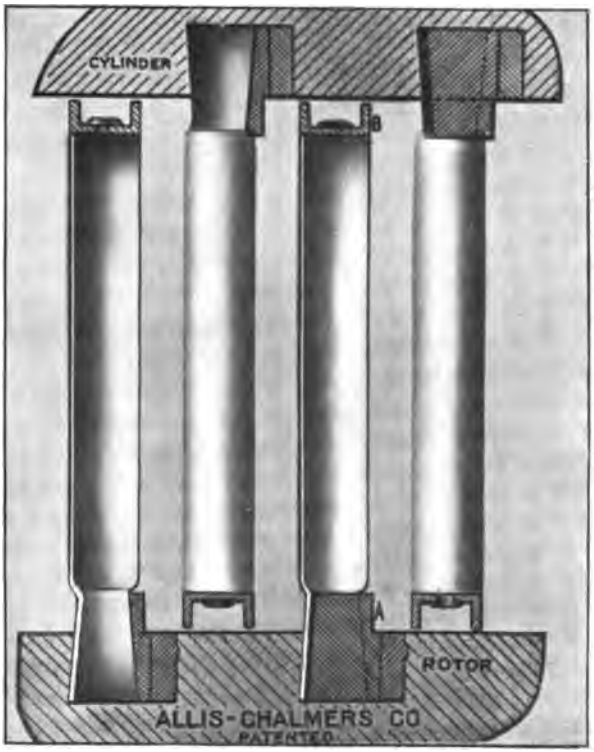
1. Channel Ring Shroud. 2. Caulking Ring.
3. Tapered Packing Section Ring.

in contact would wear and cause no serious damage to blading. The grooves in the casing or rotor for Willans & Robinson's blading are turned wider than the width of the foundation ring. Into the space left after segment is put in, a soft brass packing strip is put and is caulked into place by air hammers. The sides of the groove are dovetailed to an angle of from 4° to 6°, and after packing strip is hard caulked the whole segment is firmly held in place.

Blading of Rotors and Casings.

Rotors.—The segments are fitted into rotors by having a stop piece, as before described, put into grooves. Against this stop piece

the first segment is put, and driven up against stop; the other segments are then put in and row completed, the stop piece being taken out before putting in last segment. Two or three packing pieces in each segment are slightly caulked with a vertical caulking tool so as to prevent them from rising up or falling out; the row is completed by the last segment having three or four loose blades at the closing end; if required, one or two are taken off until the requisite length is obtained. The blades are now caulked in place, which operation is



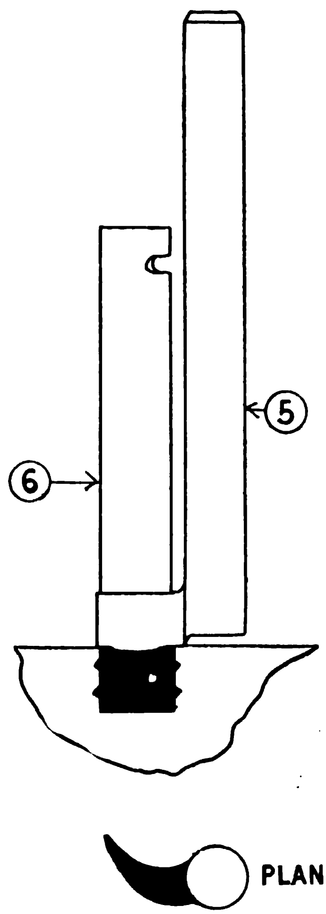
Allis-Chalmers or Willans-Robinson Type of Shrouded Blading.

NOTE.—The above system is not much in use for marine practice.

performed by vertical caulking tools and hammers of varying weight, according to the size of packing pieces which are being caulked. A portion should be caulked, and then a space missed, then another portion caulked, coming back again on that portion missed. If the caulking is carried right round it will tend to make the last segment rise out of the groove. Usually the blades are given three blows with hammer, heavier hammers being used on larger blades.

After the rotor is completely bladed it is now put into the lathe for tipping points of blades. The rotor is revolved at maxi-

mum speed of lathe, and the blade rows trued up by means of pointers fixed in line with the blade row being tipped; flat pieces of wood which go in between rows of blades are used, and blades are pressed one way and another until row is running true. It is best to true up a portion of row square from the rotor drum. This

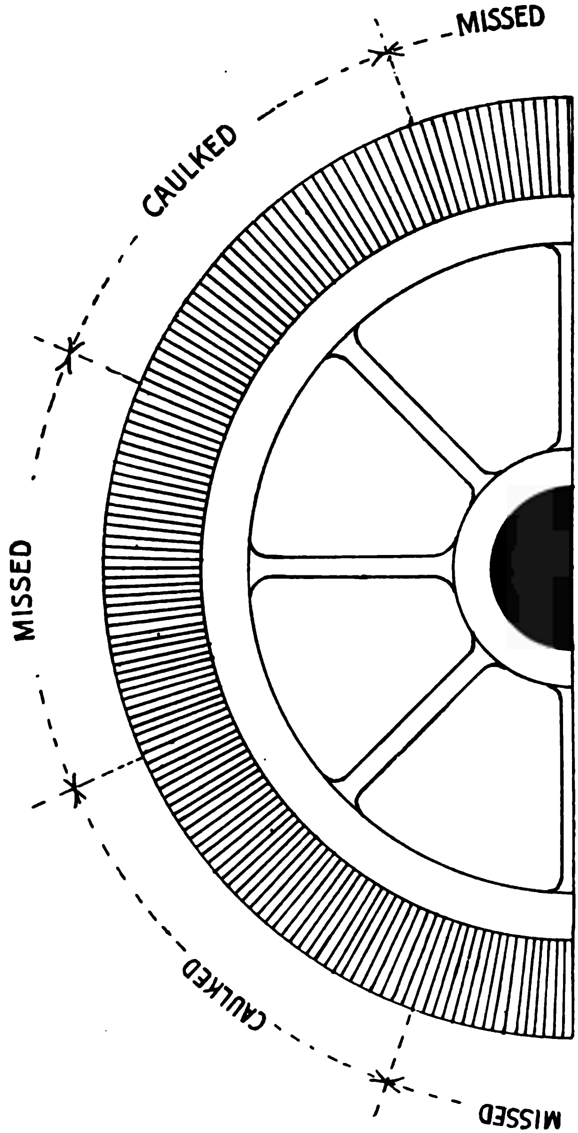


Segment Blading.

5. Caulking Tool shaped to clear Binding Wire. 6. Blade.

serves as a guide in trueing the rest of the row. The tipping is done by means of a tool having a sharp razor edge, and a very light cut is taken over tips of blades until they are reduced to the required diameter. This operation is important, as on it depends the radial clearance of the blades.

Casings.—The casings are first prepared for blading by the fitting of stopping pieces. These are made of soft brass, and their purpose

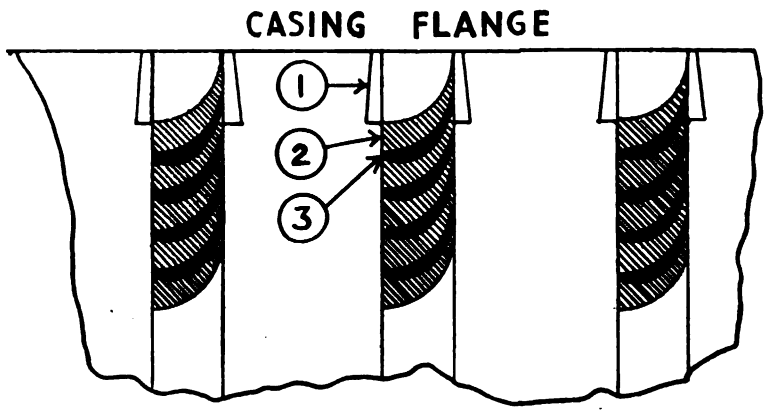


Method of Blading by "Segment" System.

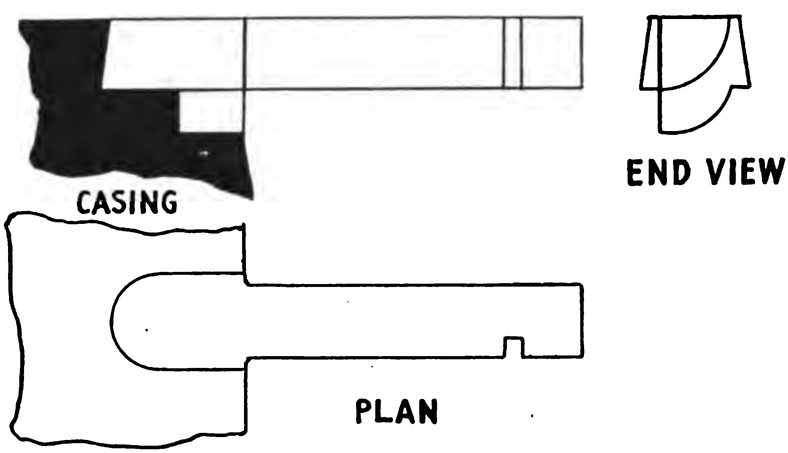
To ensure correctness of spacing and equal number of blades per row, the final caulking-in is completed as shown.

is to act as a stop for the blades at the joint of the casing. The grooves at the joint are dovetailed, and into this dovetail the stop pieces are fitted. This dovetail varies in depth and width according

to the size of blades of which the expansion is composed. After dovetails are cut, which is usually done in a milling machine, the stoppers are fitted into same, care being taken that they bear properly



View of Lower Half Casing (looking inside).
1. Blade Stopper. 2. Packing Piece. 3. Blade.

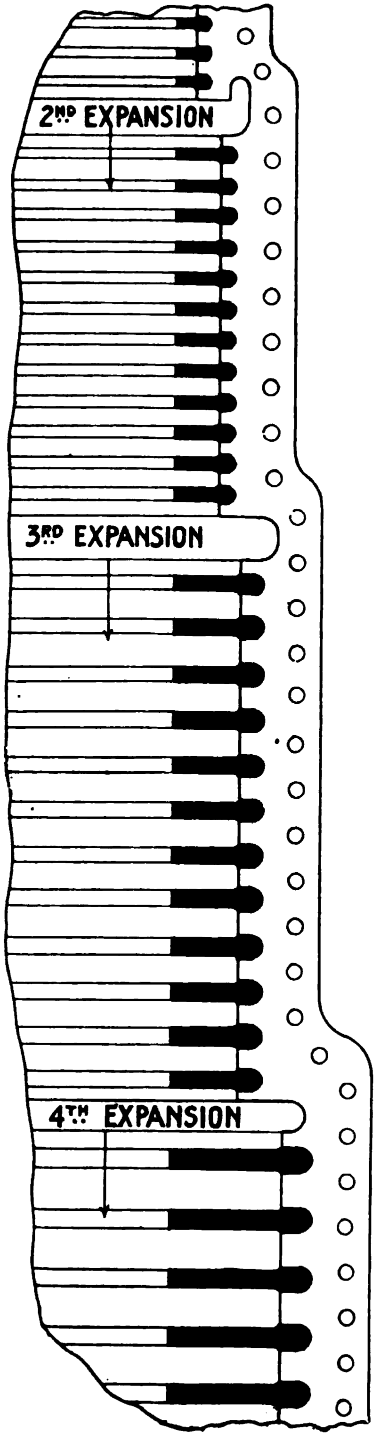


Casing Blade Stopper.

NOTE.—The stopper is fitted into the casing flanges top and bottom, and sets the angle of the blades.

Method of Blading Casings.

at the heel. One of the stoppers has a convex face of same shape as face of blade, and the other is concave, the shape of the back of the blade. The first segment is butted hard up against the stopping piece; the other segments to complete the row are butted against



Part of Turbine Casing showing Blade Stoppers in position.

this one, then the last segment is fitted in and stopper piece driven into place. At the end of the stopper piece a groove is cut in line with the binding wire on segment. The wire, which is left long for this purpose, is put into groove in stop piece and soldered in place. After all segments are fitted, the casing is now gauged across joint and stays put across casing; this is to prevent distortion of casing when caulking segments. It is usual to draw casing in to about .01 to .02 less than the finished diameter as it tends to keep the casing from distorting in the middle, to the same extent as it would if unstayed. The blades are caulked by caulking tools made so as to stand out clear of blades (see sketches). The point of the tool which strikes on packing piece is slightly rounded so as to deal the heaviest blow in the middle of packing piece. After this operation the stays binding the casing are taken off, and the casing gauged to ascertain the amount of distortion through caulking. The blades are now set up to length stick, which is set from the dummy face, and by means of wood battens the blades are squared up to same.

NOTE.—No permanent stoppers are fitted in the rotor grooves.

For good turbine efficiency results it is imperative that the angles, pitch, and position of the blades be as nearly mathematically correct as possible, any tendency to unequal circumferential pitch or angle (the blading being then known as "staggered") resulting in loss of efficiency.

The operation of "blading" being carried out entirely by hand labour, forms the most costly item in turbine construction.

The top of stopping pieces are milled flush to joint of casing, and the sides are also milled to exactly the breadth of the blades so as to ensure the same clearance all round the row. Both halves of casing having been dealt with in this manner, the horizontal joint is re-bedded, and casing is now ready for tipping of blades. The tipping is done in a horizontal boring machine by means of a tool similar to that before described in the tipping of rotors. After tipping, the blades are ragged by hand, *i.e.*, a small scraper is drawn across tips of blades so as to remove any rag or burr which may be formed on blades. It is advisable to lace the ends of segments together with copper wire so as to strengthen blades while undergoing the operation of tipping.

Difference in Number of Blades per Row.—In the blading of large rotors it sometimes happens that the number of blades per row varies by 5 or 6, owing to some of the "bladers" caulking the packing pieces further down than others, and so reducing the number of blades by increasing the space between each. As the process of blading is done entirely by hand the foregoing result is only to be expected, absolute uniformity of blading being almost impossible of attainment.

Stresses on Blades, &c.—It should be observed that the casing blades are subjected chiefly to a bending stress, due to the action of

the steam in striking the blade surfaces, whereas the rotor blades, in addition to a bending stress, are also subjected to a severe tensile stress set up by the centrifugal force of the rotor when revolving, and tending to throw out bodily the blades and packing pieces. To resist this, the grooves cut in the rotor drum and casing are serrated, as shown on page 87, to lock the packing pieces and blades in place.

The little stamped-out grooves shown near the bottom of the blades (both casing and rotor) allow of the packing piece being caulked in to lock the blades firmly in position.

Each "stopper" is of the same length as the corresponding blades. Regarding the materials employed for the blades, and clearances allowed, Mr E. M. Speakman says :—

"The material of which blades are usually made is a mixture of cheap brass containing about 16 per cent. of copper and 3 per cent. of tin. Alloys containing zinc are extremely unreliable for high temperatures, but blades containing about 98 per cent. of copper have been found very satisfactory for use with high superheats. More recently a material containing about 80 per cent. of copper and 20 per cent. of nickel has been adopted, and this is undoubtedly the best blading material existing. Steel blading, drawn in the same way as the usual brass section, has been used in the United States with fairly good results. The process of drawing turbine blades gives an extremely tough skin to the metal used, not only increasing the tensile strength, but greatly decreasing the chances of erosion.

"It seems probable that the usual caulking piece now adopted will be discarded in favour of a machine-divided strip into which the blades may be fitted, and instead of the slotting, wiring, lacing, and soldering process at the tip, a similarly machine-divided shroud will be used, giving a far stronger construction, and enabling finer clearances and better workmanship to be obtained; at the same time considerably reducing the cost of manufacture, and the risk of blade stripping.

"The chief causes of the latter may be set down to bad workmanship in fixing the blades, defective blade material, excessive cylinder distortion (this is probably the most fruitful cause, and is a serious one, being due to bad design), whipping of turbine spindles (which is also due to bad design, or bad balancing), wear of bearings (which is very remote), and the introduction of extraneous substances such as water or grit. In fact, blade stripping may be said to generally occur from preventible causes. Small vibrations of very high frequency occasionally set up an action in certain rows of responsive length that fatigues the blade material and causes the loss of blades without any fouling at all.

"Due to the action of the steam, an end thrust occurs in the direction of the propeller, which is advantageously used in partially balancing the propeller thrust, thereby reducing the size of the thrust block necessary. A margin must be allowed here, as the propeller thrust is not entirely balanced by the pressure on the annulus between the dummy-ring diameter D and the spindle, Fig. 5, C, plus the end pressure on the blades. For the diameter D to give the required annulus, as well as that of the propeller, the effective thrust must be carefully calculated; and experience shows that there is a drop in steam pressure varying from 10 to 15 lbs. per square

inch between the pipe inlet to the H.P. receiver and the first row of blades, which should be considered in designing this balancing area. The number of rows of dummy packing used varies according to the designer's judgment very largely, and may be modified according to the pressure and the clearance allowed—say a 7-1000th to a 15-1000th of an inch in electrical work, and rather more in marine work.

"The dimensions of the astern turbine are arrived at in the same manner as those of the ahead, the efficiency being largely sacrificed on account of weight and space; generally, the mean diameter is made practically the same as that of the H.P. drum.

"To a large extent, the inferior manœuvring capabilities of the earlier turbine steamers were due to insufficient astern power.

"It may be remembered that in a marine turbine the spindle is in compression and the cylinder in tension when working. In electrical turbines where the end thrust must be eliminated by the use of balancing pistons, the spindle is in tension and the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff enough, as well as strong enough, for any sag in the spindle will destroy the clearance. The stresses due to centrifugal force are very low in the Parsons turbine, and except in occasional L.P. barrels do not exceed about 7,500 lbs. per square inch, while at the H.P. end they are usually under 2,000.

"The pressure on the bearings in a turbine is only due to the weight of the spindle, plus the negligible addition in marine work of that due to any gyroscopic action; it may be taken as from 80 to 90 lbs. per square inch as long as the rubbing velocity does not exceed 30 feet per second. If it does, the pressure must be reduced so that the product of pressure × velocity does not exceed 2,500-2,700. In land work, 50 lbs. × 50 feet is very common. The friction heat of the bearings added to that due to conduction through the pedestals necessitates the use of large oil coolers, and in the case of very high temperatures, of special kinds of oil. If possible, the bearing temperature should not exceed from 140° to 150° F., though the writer has known of 190° F. being used without trouble. In marine turbines this temperature is usually much lower. Rigid bearings are used for marine spindles, not the flexible type adopted in land work.

"Space does not permit of more than passing reference to cylinders; but it would be difficult to exaggerate the importance of very careful design in this connection. Cylinders, with heavy flanges on the centre line, distort in a very curious fashion when heated with their axis horizontal, and measurements taken off a hot cylinder on a surface plate with micrometer gauges reveal some very remarkable facts. When working, the temperature along the cylinder falls possibly from 400° to 100° F. in a distance of 6 or 8 feet, and, unlike the reciprocating engine, this remains constant; the radial expansion is consequently more at one end than the other; while at any point along the turbine the tendency is to expand less at the flanges than at the top and bottom. For this reason ample clearance must be allowed; exactly what this will be when spindle and cylinder are hot is hard to say, but it seems most likely that the total clearance area will differ but little from what it is when cold.

"The longitudinal expansion when hot is often very marked, and in all turbines necessitates provision for the resultant movement at one end. In marine work the after end of the cylinder is secured to the vessel, the engine

seating also performing the function of a thrust-block seat, while the forward end slides forward, taking with it the entire shafting. The thrust block is at the forward end of the cylinder, and also performs the duties of an adjustment block for setting the longitudinal clearances, to do which generally necessitates uncoupling the shafting abaft the turbine.

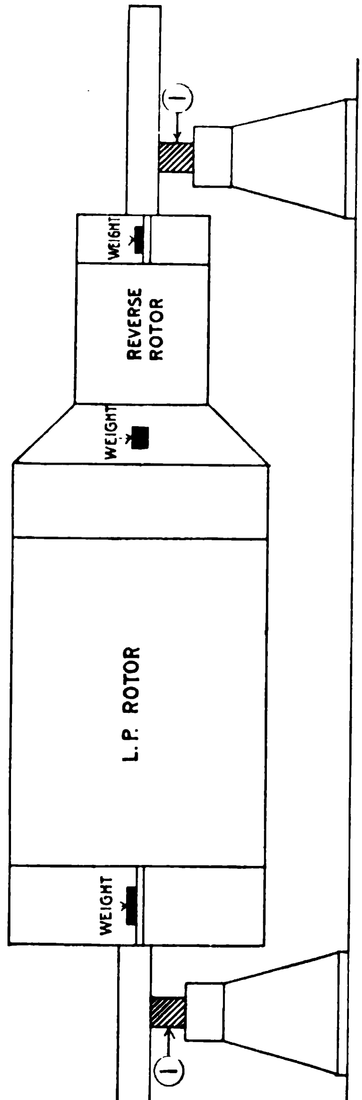
"The difference in expansion between the cylinder and spindle, from the thrust block to the dummy ring, may be the cause of serious difficulties in large marine turbines, unless the closest attention is paid to this feature in the design; and 'warming up' with these large cylinders needs possibly even more care than is essential with large piston engines.

"On shipboard, the turbine cylinders are practically under one's feet, and the radiation from them is very unpleasant, especially if there is any leakage from the glands. To all who are responsible for the lagging of cylinders and the system of ventilation in turbine engine-rooms I would call attention to the possibility of their having to stand a watch of four or six hours on the top of the H.P. cylinder, such as is the case in the 'Eden' or 'Amethyst,' the heat in the latter vessel being almost unbearable. With reciprocating engines, one stands on a comparatively cool lower platform with the cylinders overhead, and with some chance of the hot gases rising clear, but in naval turbine work under a low deck this point has not met with adequate attention.

"In the course of operation, more especially in marine work where no superheaters are used, there is a distinct tendency for the turbine to be supplied with wet steam, the effect of which on the economy is very marked. Experiments that have been made show that the percentage increase in consumption is about twice that of the moisture in the steam. For instance, with 2 per cent. of moisture in the steam at the first row, the consumption is increased about 4 per cent."

Balancing of Rotors.—Two balances are required to ensure a steady running rotor: these are the static balance and the dynamic balance. The static balance is obtained by revolving the rotor by hand on a pair of knife edges or balancing stools, which consist of two supports of cast iron, having on top a truly machined face of hard grained cast iron. This face should be 2 to 3 in. in breadth, according to weight of rotors which are likely to be balanced. The stools are levelled up longitudinal, and to each other, after which the rotor is laid on these, resting on the bearing part of the shaft. The rotor is revolved and allowed to come to rest, and the point at the bottom will be the heavy part. Temporary plates are fixed on to the wheel opposite this, and the rotor revolved and weights added until it will stand at rest in any position. This being arrived at, the next operation is to determine the distribution of weights, which is a most important part, as on this may depend the success of the static balance in relation to the dynamic balance. If it is an H.P. rotor, having two wheels of the same diameter—one forward and one aft—it is usually best to put one-half of weight on each wheel: if, however, an L.P. and astern rotor combined is being balanced, having three wheels—two of the same diameter, and the after one smaller—it is advisable to put a slightly heavier weight on the after wheel than on the other

two, as owing to the smaller diameter of the aft wheel the weight will be acting at a smaller leverage. An important point is to put all weights in line if possible, and to have them duplicate as regards bolt

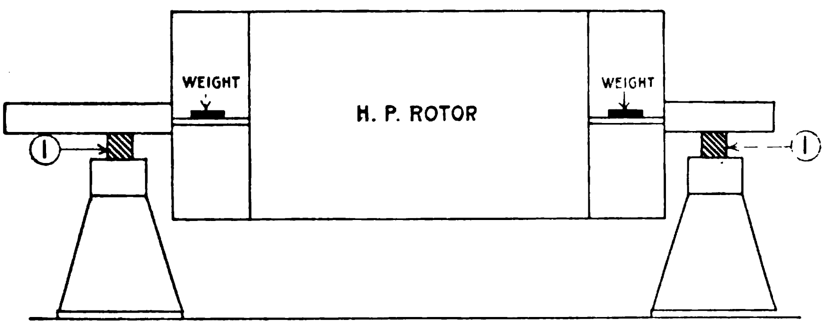


Static Balance of L.P. and Reverse Rotors.
1. Knife Edge.

holes. After weights are fitted the rotor is now tested for balance. This is done by dividing the circumference of the wheel into eight equal parts. A stationary pointer is attached to an upright batten fixed to a small knee plate which rests upon the floor, a line is drawn

across the rim of the wheel in line with the centre of the shaft, a small weight of, say, 1 lb. or so is suspended by a hook attached to the wheel; the rotor will move a small amount and then come to rest, when another line is drawn across rim of wheel: this distance is measured and note taken of same. The same operation is performed at the other seven points, and the result compared: if the rotor is truly balanced the movement should be the same at each point. If movement varies then weights will require to be altered to suit.

"**Dynamic" Balancing of Rotors.**—The turbine is balanced dynamically by one of two methods—(1) By being revolved by steam acting as under working conditions, and tested for vibration by fixed indicating pointers or pencils. (2) By some suitable arrangement of spring bearings in which the rotor is placed and rotated by some external power, such as an electric motor coupled up to the spindle, and as before a pointer indicates the vibrations on a piece of diagram paper by a more or less "waving" line. The latter method is the



Static Balance of H.P. Rotor.
1. Knife Edge.

most accurate, as the rotating force acts from the centre outwards, and, if anything, tends to magnify any lack of balance or difference in weight which the rotor may possess. At the same time, few rotors are actually in a state of perfect balance, as no really scientific and accurate method of dynamic balancing has yet been devised for turbines. Thin plates are either pinned on as required to those portions of the rotor which are found to be lighter and therefore require added weight, or metal is chipped off the "balance" strips on the heavier positions.

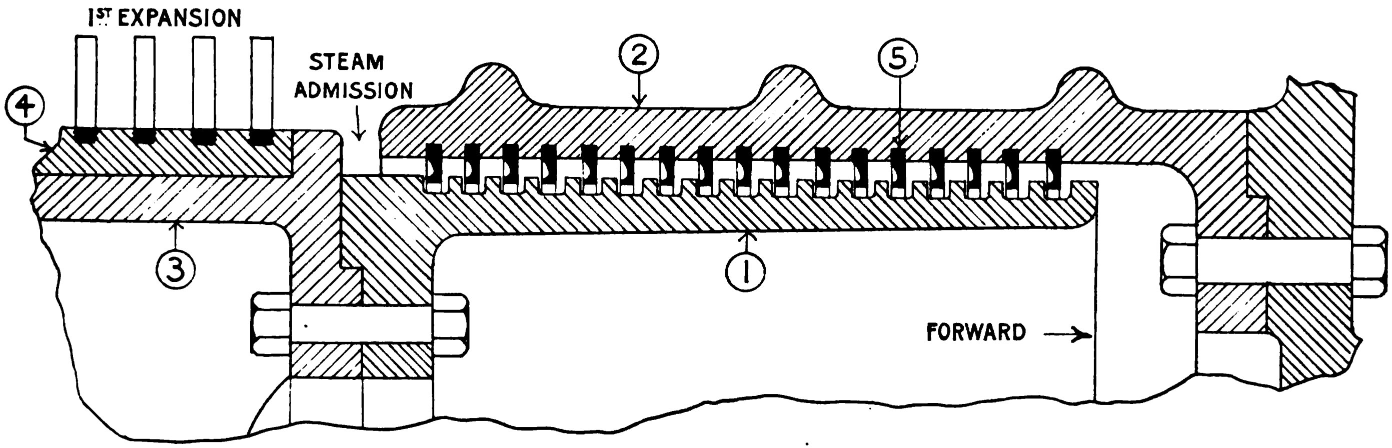
The rotor should be revolved to at least 20 per cent. above the required revolutions. Should the balance be imperfect it will show by vibrations being set up, and if the balance is very bad may result in damaging the blading of the turbine. When the rotor is at its worst point of vibration, marks should be put on shaft at forward and after end: it is important that these marks should be put on at the same time. The turbine is now stopped and the relation of the marks to each other examined. If the marks on each end are opposite, or not in line with each other, and one end in line with the weights and the other

opposite, then by taking a weight from one end and putting it on the other this will ensure the static balance being maintained and the dynamic balance improved, if not perfected. After balancing, the bolts through the weights are riveted over on the points to prevent slackening back. In running turbines under steam, especially when developing high speed, it is found that there is usually a point when vibration will begin to show—this is termed the critical point of speed. This is believed to be due to a change in the axis of rotation caused through deflection, or bending of the shaft, which causes the rotor to revolve round a new axis, termed the axis of deflection, and which alters the centre of gravity of the rotor and causes vibration. This may not occur at the highest speed, as the vibration sometimes decreases as speed is increased.

The various elements which go to make up the complete rotor such as drum, wheels, and spindles, are each carefully balanced before assembling, then the whole is again statically balanced on knife edges, after which an approximate dynamic balance test is employed by running the turbine under steam and noting the vibrations at either end.

The rolling balance is tested by noting if the rotor will stop dead at any position when moved by hand along the knife edges. If not, the balance may be first adjusted by sticking on pieces of clay at the lighter positions, and when the balance is thus corrected, the clay is replaced by metal plates which are pinned on to the wheels. With some firms the practice is to fit the wheels with eight balance strips which are cut off as required to lighten the heavy side of the rotor, in place of adding weights to the lighter side. The extra weight of the balance strips may amount to about 30 lbs. in fair sized rotors, and may be thus reduced to suit the balance as required. When testing the dynamic balance by running under steam, it is advisable to leave the static balance undisturbed, and this is arranged for by merely taking weight from one end and giving it to the other end, keeping of course the same circumferential position. When the dynamic balance is out, it is indicated by vibrations at the heavy end, and, as before mentioned, this is corrected by transferring weight from one wheel to the other. No system of calculation is employed in either balances, and the excellent results generally obtained are entirely the result of experience in the work together with good workmanship.

Dummies.—The dummies are placed at the steam admission end of each turbine, and two kinds of dummies known respectively as radial and facial are employed: the facial dummy is usually fitted in the ahead turbine, and the radial dummy in the astern turbine. The principle of the dummy is to prevent the steam from escaping through the interior of the rotor to the exhaust end of the casing instead of doing its legitimate work in passing through the blades of the turbine. Another reason is that if no dummies were fitted, the full initial pressure would be on the glands instead of exhaust or terminal pressure. A facial dummy consists of two parts called the casing dummy and the rotor

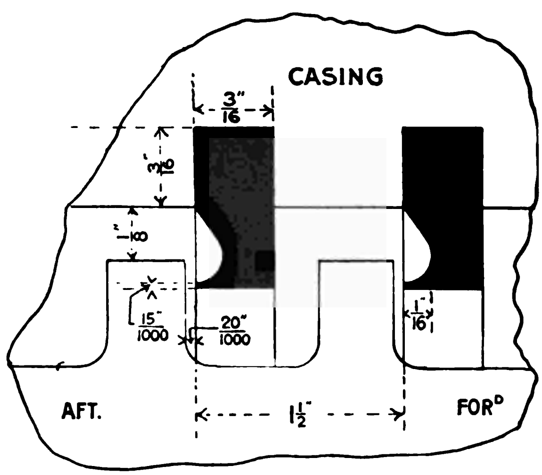


Ahead Type of Dummy (Facial).

1. Rotor Dummy.
2. Casing Dummy.
3. Wheel Drum.
4. Rotor Drum.
5. Undercut Brass Rings bedded into Casing.

The difference in diameter between the rotor dummy and drum, as shown, forms the annulus upon which the steam acts to partly counterbalance the propeller thrust.

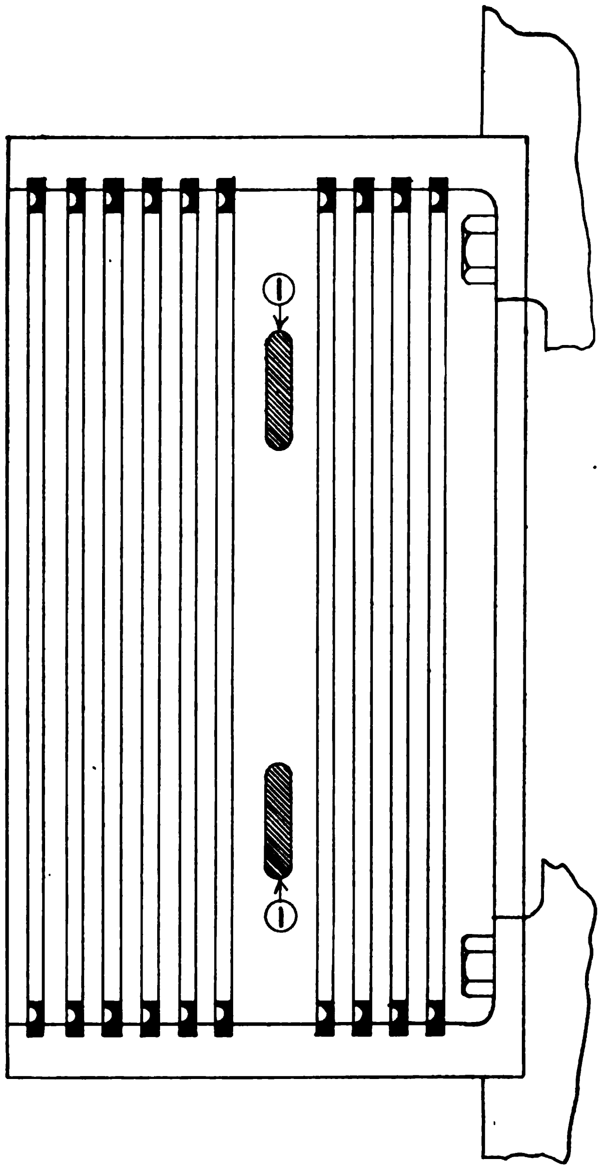
dummy. The casing dummy is a cast-iron cylinder, which is in two halves, bolted together at the horizontal joint. This cylinder is bored out and grooved, the grooves being usually $\frac{1}{8}$ in. wide and $\frac{3}{16}$ in. deep, and into these grooves are driven brass strips. The brass strips are bent to the radius of the cylinder, and a serration made in them, so that the serration is just flush when the strip is driven into groove. After the blades are in place, the metal of the cylinder is caulked into the serration, thus binding the strips. The strips are cut in lengths of 6 in., and at each end of the half of the casing in one row a 9-in. piece is put in, and in the succeeding row a $4\frac{1}{2}$-in. piece is put in, so that the joints in each row are not in line. The strips are left ".012" clear of each other at the ends, so as to allow for expansion. After the blades are all in place and caulked, the dummy is put into lathe, and the blades turned up.



Ahead Dummy Facial Rings.
(With average dimensions.)

The blades have a face bearing of ".015" so as to ensure that if the rotor dummy should touch, the friction caused thereby would be reduced to a minimum. The rotor dummy is of steel, and usually is a round cylinder, although sometimes it is made in two halves. The dummy is rigidly bolted to the rotor, and turned up in place. A series of grooves corresponding to the brass strips in the casing dummy are turned out, having a fillet in both sides of groove, the grooves being $\frac{3}{16}$ in. deep. The brass strips in the casing dummy project into the groove in the rotor dummy $\frac{1}{8}$ in. When the rotors are set to position in the casing, the factor which determines this position is the dummy clearance, this varying according to the size of turbines. For average sizes the clearance is usually as follows:—

.015 to .020 in the high-pressure turbine,
and .020 to .03 in the low-pressure turbine.

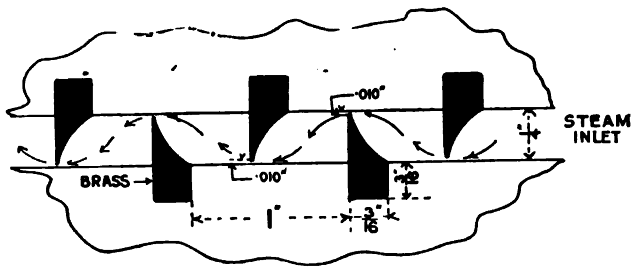


Ahead Dummy Casing.

1, 1.—"Leak-off" Ports.

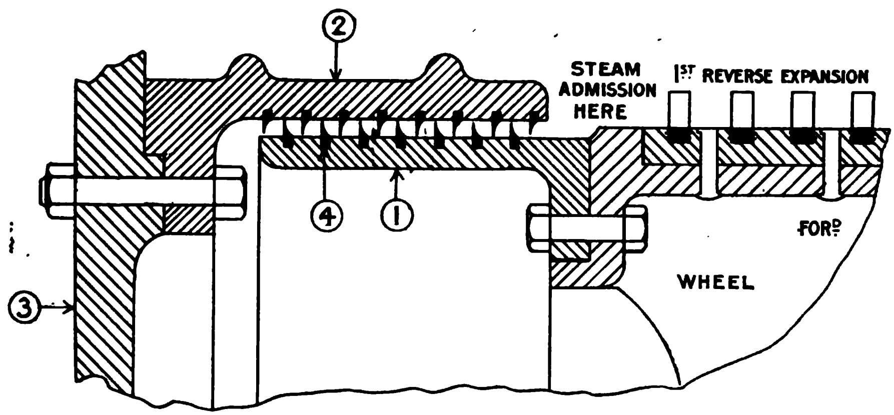
Note.—Usually not less than 20 of the small brass rings shown are fitted to each ahead dummy. The rings are formed of $\frac{3}{16}$-inch brass at a pitch of about $\frac{3}{8}$ inch. The "leak-off" ports shown are omitted in later designs.

Radial dummies are generally fitted in the astern turbine, and as in most turbines the astern rotor is part of the L.P. rotor, the expansion under steam takes place in an aft direction; it will thus be readily seen that a face dummy would not be possible. A radial dummy consists of two parts, the casing and rotor dummies. The casing dummy is a cast-iron cylinder bolted to the end of the turbine casing, and has grooves cut the same as a facial dummy; the grooves are



Radial Fins of Astern Dummy.
(With average sizes.)

bladed in the same manner with brass strips, and the strips are turned up to a knife edge, with one straight side, and the other side with a fillet as shown in sketch. The rotor dummy is also grooved, and in this case brass blades are also put in; these blades project



Astern Radial Dummy.

1. Rotor Dummy. 3. End of Casing.
2. Casing Dummy. 4. Radial Fin Rings.

NOTE.—The number of rings usually fitted varies from 6 to 8.

in between the blades on the casing dummy, but do not touch, and the clearance allowed from the tip of blades is usually .015 to .020. The steam escaping through the blades is first wire drawn, and then expanded, and so on until it escapes at the end into the interior of

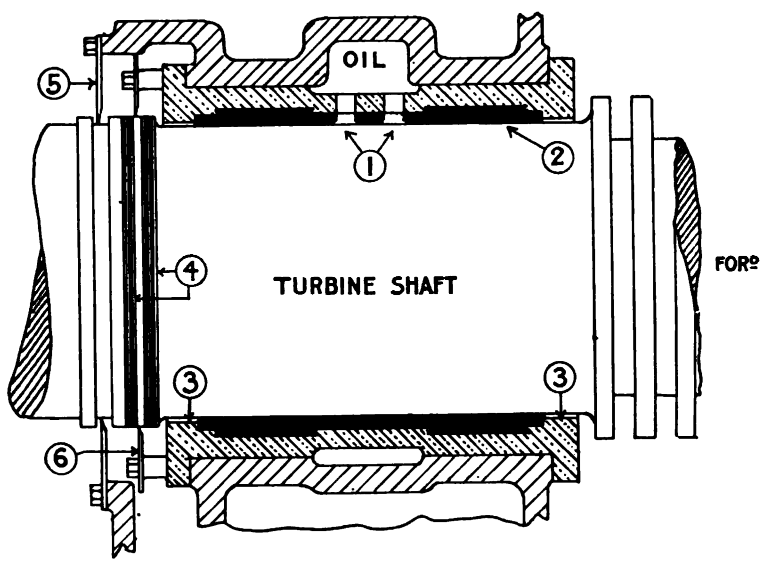
the rotor, which is in a vacuum if a low-pressure astern dummy and to the exhaust pipe leading to the L.P. if a high-pressure astern dummy.

Dummy "Leak-off."—The ahead dummies are sometimes supplied with a "leak-off" from the space between the sets of rings to the 3rd expansion of the same turbine.

Any steam which finds its way past the inward set of rings passes away by the "leak-off" ports and pipe to one of the other expansions where the pressure is less (see sketch).

As before stated, the "dummies" prevent steam leakage at the high-pressure ends of the rotor and act so as to produce (by means of wire drawing off the escaping steam) a "water seal" by the resulting condensation.

Main Bearings and Adjusting Blocks.

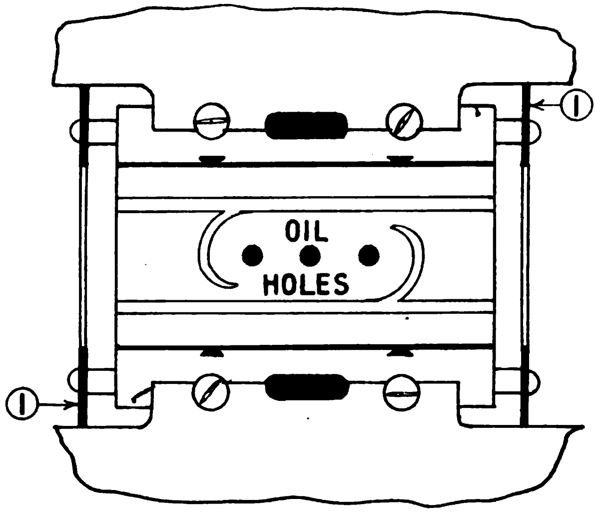


Plan of Main Bearing.

1. Oil Holes. 2. White Metal. 3. "Reliefs," $\frac{15}{1000}$ in. clear, to prevent damage to blades if white metal runs out. 4. Oil Baffle Grooves. 5. Oil Baffle Plates.

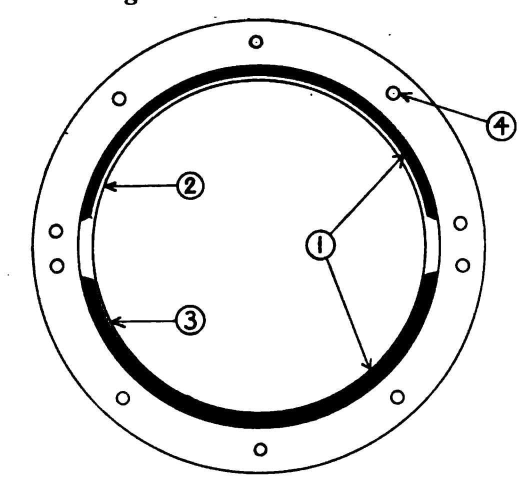
Main Bearings.—A bearing is fitted at each end of each rotor, and is of the usual marine type, being of either brass or cast iron (the former for naval work), filled up with white metal. The bearings are held in place by four large screws, each one being partly in the seat and partly in the brass as shown. Small holes are bored through the top or bottom half of the brass, and the oil is forced through the oil holes by the oil pumps, as elsewhere described. The pressure on the bearings is usually from 70 to 80 lbs. per sq. in.

Oil Baffles or Deflectors.—Ring plates of brass in two halves are secured to each end of the after brass, and to the after end of the



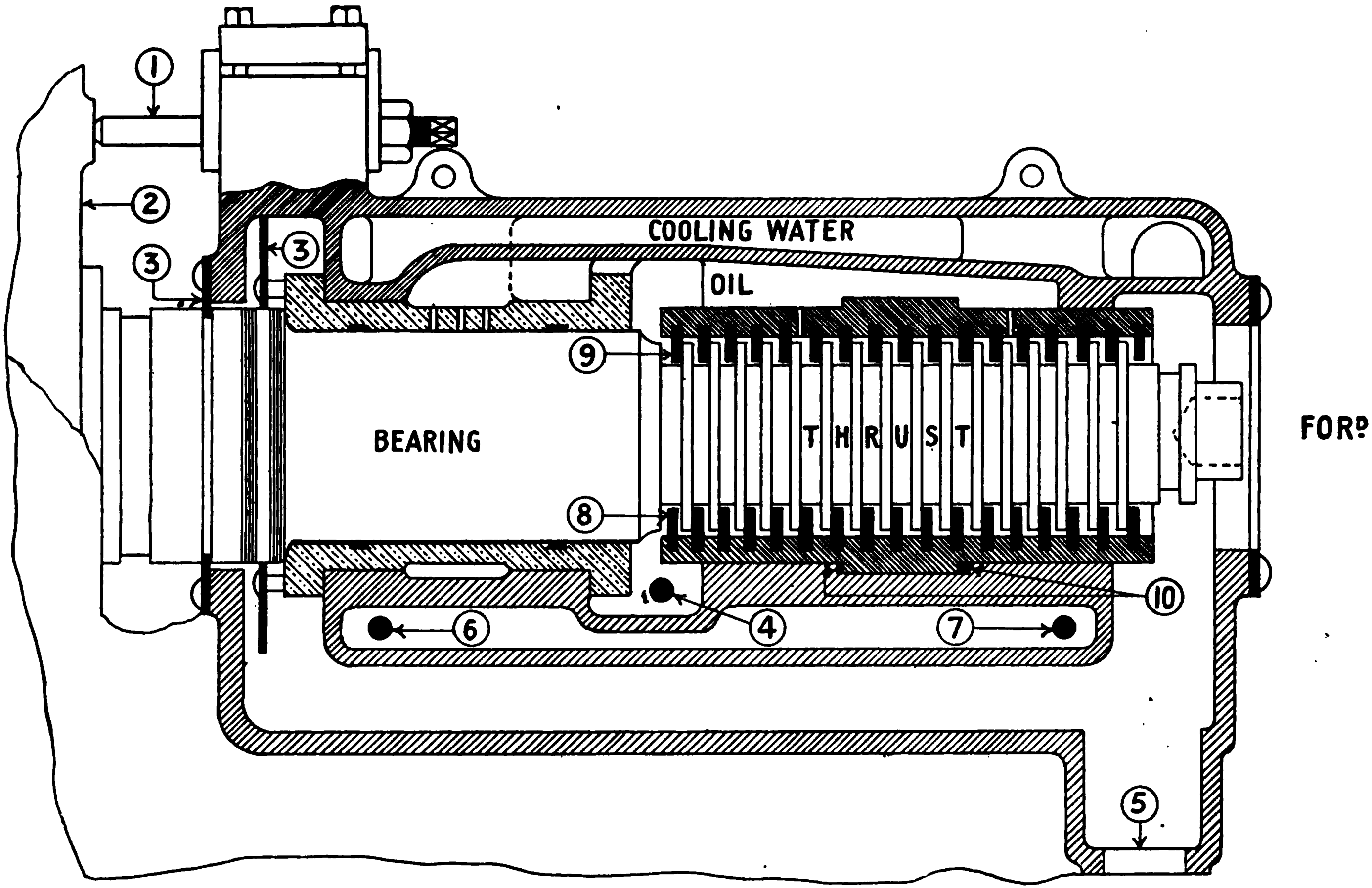
Lower Half Main Bearing.
1, 1. Oil Deflecting Rings.

forward brass, to act as oil baffles, that is, to prevent the ingress of oil to the rotor casing and blades. These baffle ring plates fit in



End View of Main Bearing Bush.
1. White Metal. 2. Clearance of .015 in. 3. Clearance of .004 in. 4. Baffle Plate Studs.

very closely round the spindle circumference and thus limit the oil to the required positions. These are usually of $\frac{1}{16}$-in. or $\frac{1}{8}$-in. brass plate, the plates being knife edged and fitted .004 clear of shaft. The

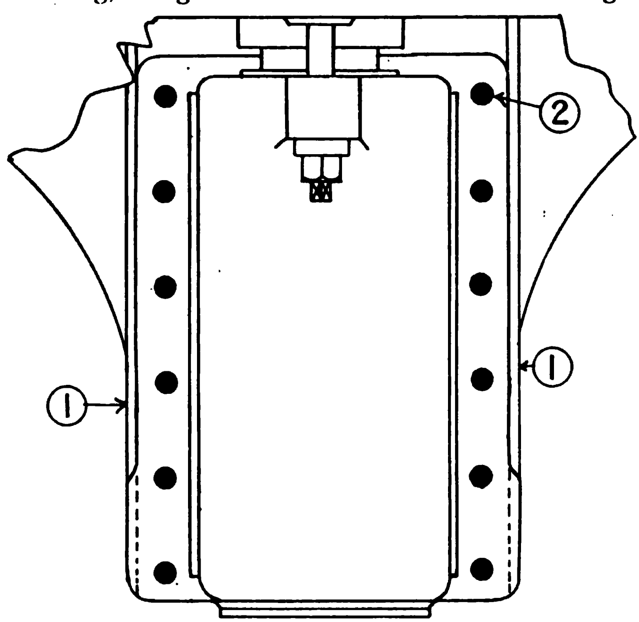


Sectional View of Forward Bearing and Thrust.

1. Adjustment Stud. 2. End of Turbine Casing. 3. Oil Deflecting Rings. 4. Oil Inlet (from oil pumps). 5. Oil Drain to cooling tanks. 6. Cooling Water Inlet. 7. Cooling Water Outlet. 8. Ahead Thrust Half Rings. 9. Astern Thrust Half Rings. 10. Thrust Adjustment Half Rings.

top half of the bearing is also lined with white metal, and is usually left .015 clear of shaft to allow for passage of lubricating oil, and also to allow for expansion of shaft when heated up. It is also advisable to have bearings .008 wider in diameter across the sides for the same purpose.

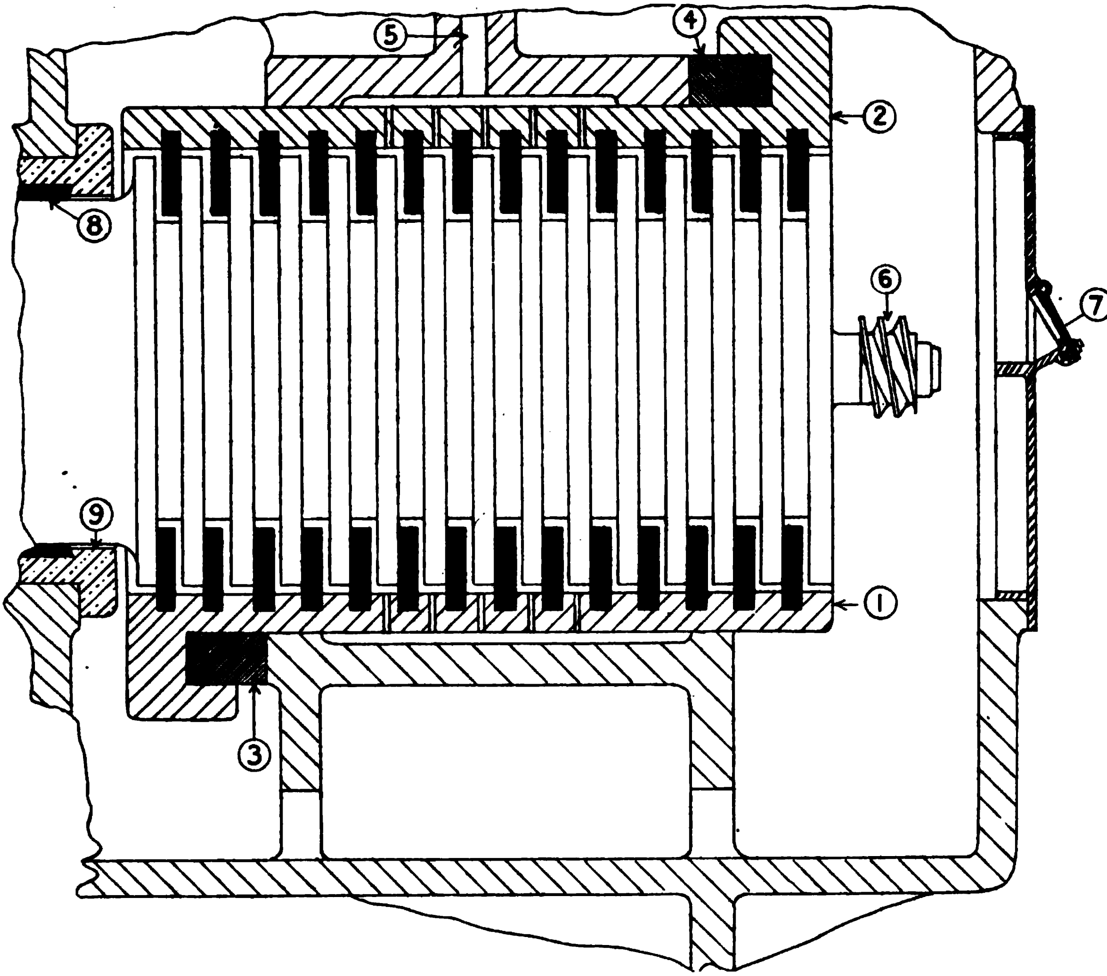
Thrust Block.—The adjusting block in the Parsons turbine consists of a forged steel brush grooved out for receiving brass rings. The thrust usually consists of seventeen rings. The rings are fitted into place, and the steel is caulked into a serration on the rings. Through the rings, with the exception of the two end ones, holes are bored, usually three in each ring, and gutters are cut from these to the edge.



Plan of Thrust Cover.

1. Guides for Cover. 2. Bolt holes, larger than bolts to allow for adjustment.

The reason for not boring the two end rings is to prevent the oil from having a too free escape. The thrust bush consists of two parts, top and bottom, the rings in the bottom bush bearing against the forward side of the collars on the shaft, and the rings in the top bush bearing against the after side of the collars on the shaft. The collars on the shaft are usually $1\frac{1}{8}$ in. apart, and the rings in the bush are 1 in. thick, so that there is a space of $\frac{1}{8}$ in. between the forward and aft side of the collars. The rings on the thrust bush are bedded up on the collars on the shaft, so as to ensure that all the rings are bearing. Some firms grind the adjusting bush up against the collars on the shaft by means of ground emery and oil. When the rotor dummy is set to the requisite clearance, the bottom half of the thrust bush is

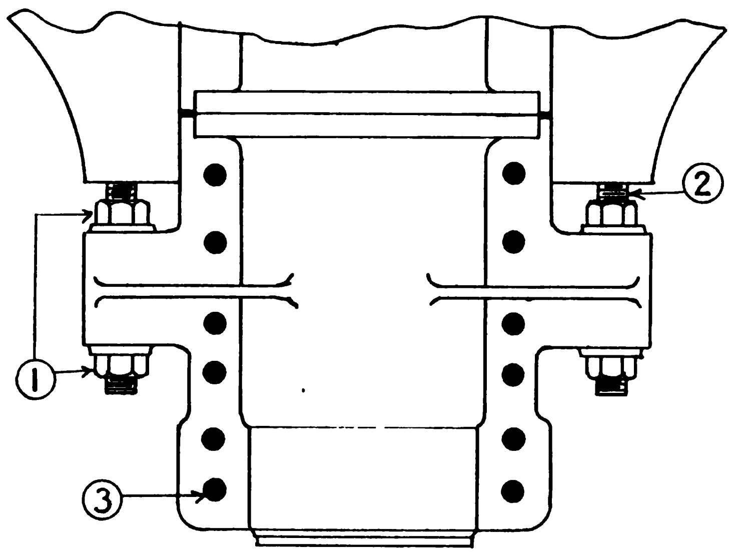


Turbine Thrust Block.

Lower Half for Ahead Thrust; Upper Half for Astern Thrust.

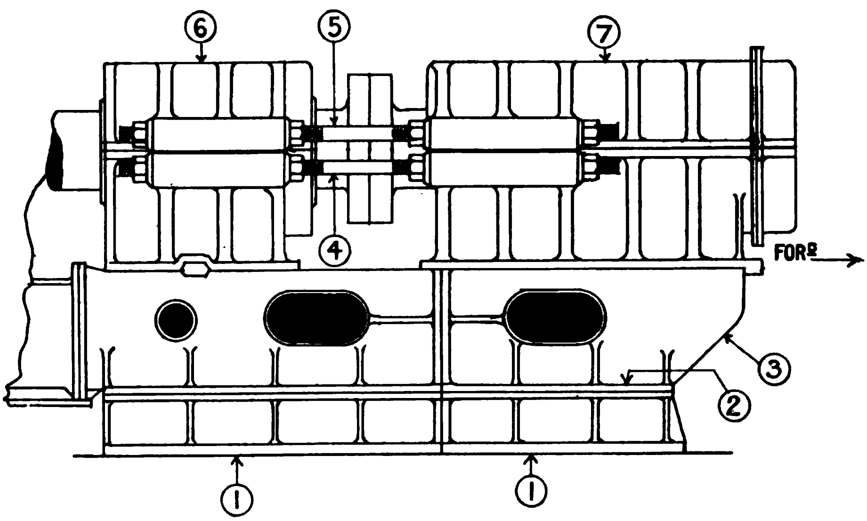
(1) Ahead thrust.
(2) Astern thrust.
(3) Taper key for adjustment of lower half.
(4) Taper key for adjustment of upper half.
(5) Oil inlet.
(6) Counter gear worm.
(7) Inspection door.
(8) White metal of main bearing.
(9) "Reliefs" for wear down.

[*To face page* 108.



Method of Thrust Adjustment.

1. Adjusting nuts. 2. Adjusting studs bearing against face on casing end. 3. Bolt holes, larger than bolts to allow of adjustment.

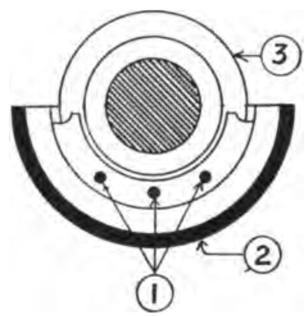


Main Bearing, Adjusting Gear, and Thrust Seating of Large Turbines.

1. Rigid seating. 2. Bolt holes, larger than bolts to allow of expansion. 3. Sliding seat of thrust. 4. Adjusting studs for lower half thrust. 5. Adjusting studs for upper half thrust. Main bearing cover. 7. Thrust cover.

put hard up against the forward side of the shaft collars, and in a space at forward side of the check on the outside of the bush a brass liner is fitted. The top cover, with bush in place, is then put on and brought forward so that rings in top half bear against aft side of shaft collars. It is then pushed or drawn aft until there is a clearance between the rings in top half of bush and collars on shaft of .004 in. to .015 in. This clearance is to allow for lubrication, and also for expansion of shaft when heated up. In average size turbines the gear usually fitted for moving the thrust bush consists of a screw fitted into a brass nut on the cover. The point of the screw spindle bears against a boss on the top half of the casing. By screwing up spindle, top half of bush may be moved forward, but the bottom half of bush can only be moved forward or aft by reducing or increasing the liner thickness in forward side of check on bush (see sketch).

It is interesting to note that the wear on the block is practically *nil*, as the propeller thrust is balanced by the pressure of steam on the



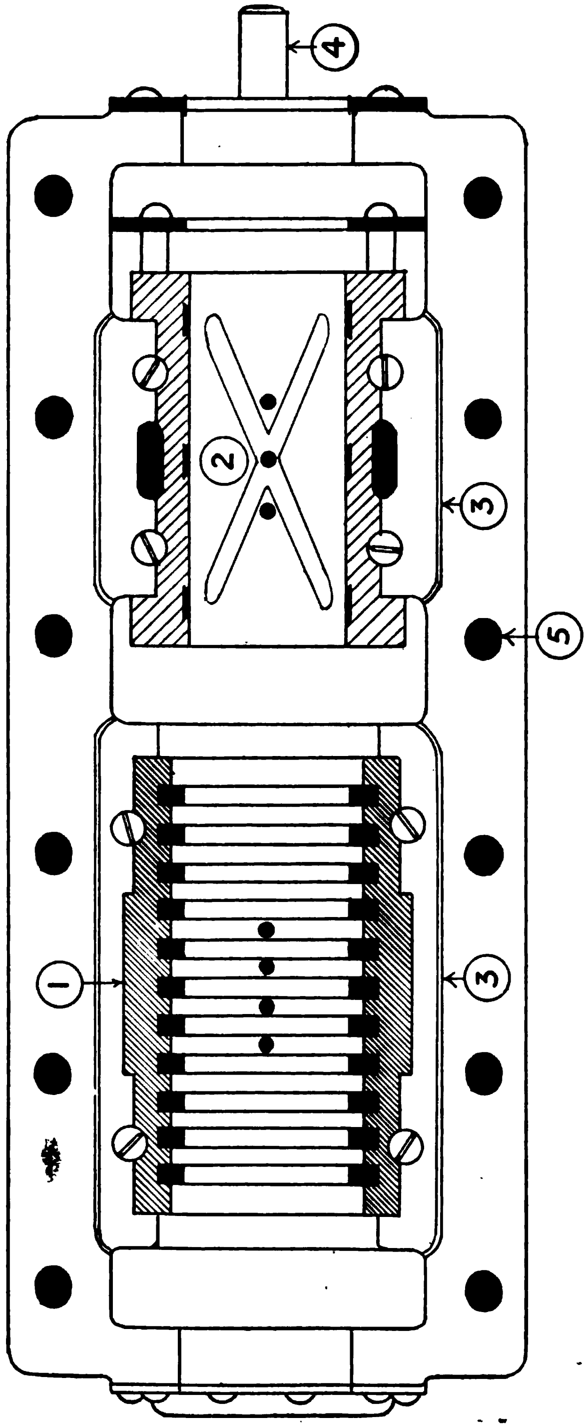
Forward End View of Thrust Block.

1. Oil Holes in Brass Thrust Ring. 2. Adjustment Half Ring. 3. Shaft Collar.

vanes acting in the opposite direction, so that to all intents and purposes the block is not really called upon to receive the thrust as usually understood in connection with ordinary marine engines. The block is chiefly required to take the thrust when steam is turned on or off.

NOTE.—Sometimes the wear takes place *aft* instead of *forward* (usually in the H.P.) owing to the blade steam thrust being in that direction.

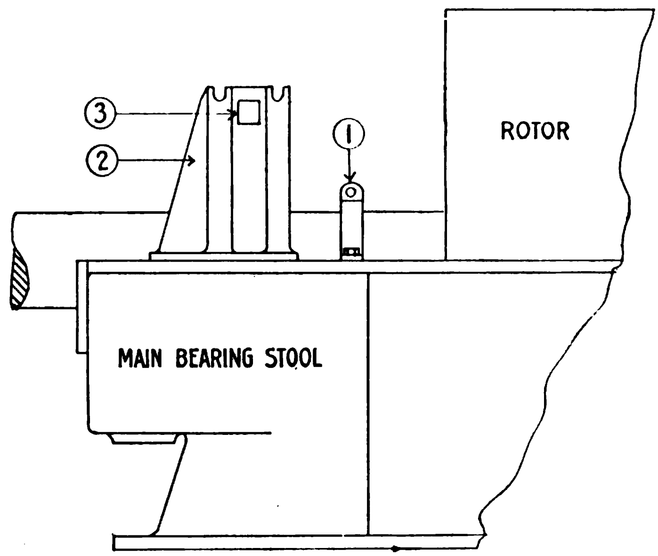
To adjust Dummy Clearance.—(1) By means of the dogs and "screwing-up" bolt forward, bring the rotor up until the dummy rings and grooves are in actual contact; next measure and note the clearance in thousandths between the after side of the finger plate and the spindle groove when in this position, and screw the rotor back until the clearance between the spindle groove and finger plate is *increased* by say $\frac{30}{1000}$ of an inch, or .03. The lower half or ahead section may then be locked in position by fitting in at the forward end or at both ends the adjusting rings.



Inside View of Thrust and Main Bearing Cover.

1. Astern Half Thrust. 2. Oil Holes in Brass. 3. Oil Gutters. 4. Adjustment Stud. 5. Oval Holes to allow of adjustment.

(2) The upper or astern portion of the thrust is set as follows:—Ease back the nuts of the cover studs, and screw round the adjusting bolt in the cover until the rings and collars come into metallic contact by the cover moving forward. Now screw down tight the cover stud nuts, and adjust the stud until when tested by "feelers" the desired clearance of say $\frac{6''}{1000}$, or .006, of an inch exists between the stud and the end of the casing; then ease back the cover nuts, and tap up aft the cover until the point of the adjusting stud and the turbine casing are in actual contact, which will thus give the determined thrust clearance; the cover bolts are then screwed down hard. It will be noted that the total clearance for oil is $\frac{6''}{1000}$, or .006. The dummy clearance if when cold is, say, $\frac{30}{1000}$ of an inch, or .03, usually decreases to $\frac{20}{1000}$, or



Rotor Guide Brackets and Lifting Gear.

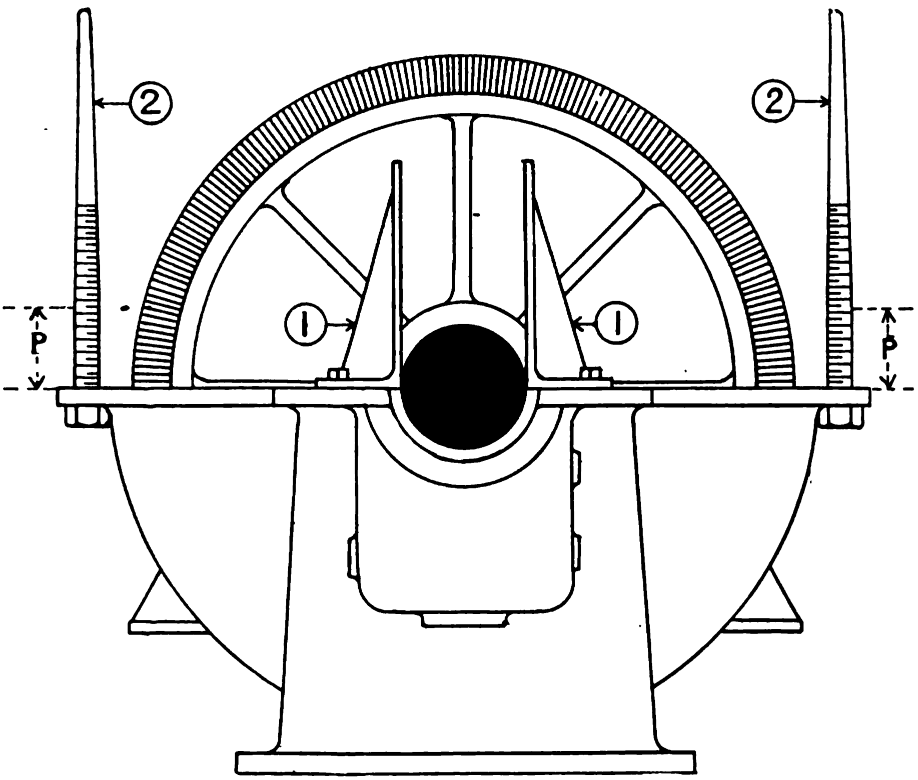
1. Lifting Strap. 2. Guide Bracket. 3. Slot for Carrying Bar.

.02, when expanded after heating up to working conditions. The engineers are therefore instructed to test the dummy clearance with the feeler gauge both previous to heating up and after heating up.

Some chief engineers test and note the clearance referred to every two or three days, or even oftener, this clearance constituting the most important and most delicate adjustment in the whole turbine, and one on which the economy and mechanical efficiency greatly depends. Should the dummy rings and grooves overstep the $\frac{20}{1000}$ or so clearance, breakdown is sure to occur by the stripping out of the small brass dummy rings; other damage may also result.

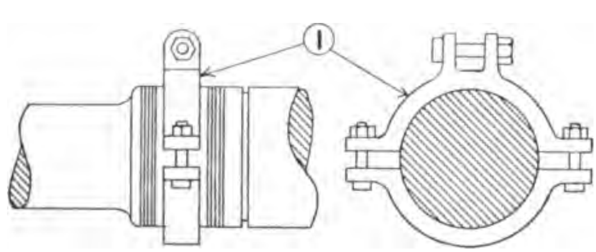
Bedding of Rotors.—The rotors being now balanced and blades tipped, the next operation is the bedding down of same. The rotors

are guided into position by means of guide columns at forward and aft end of turbine. These columns are set so that the rotor dummy will



End View of Turbine with Cover Lifted.

1. Guide Brackets for Raising of Rotor. 2. Guide Columns for Removal of Cover.
P. Parallel Portion of Columns.

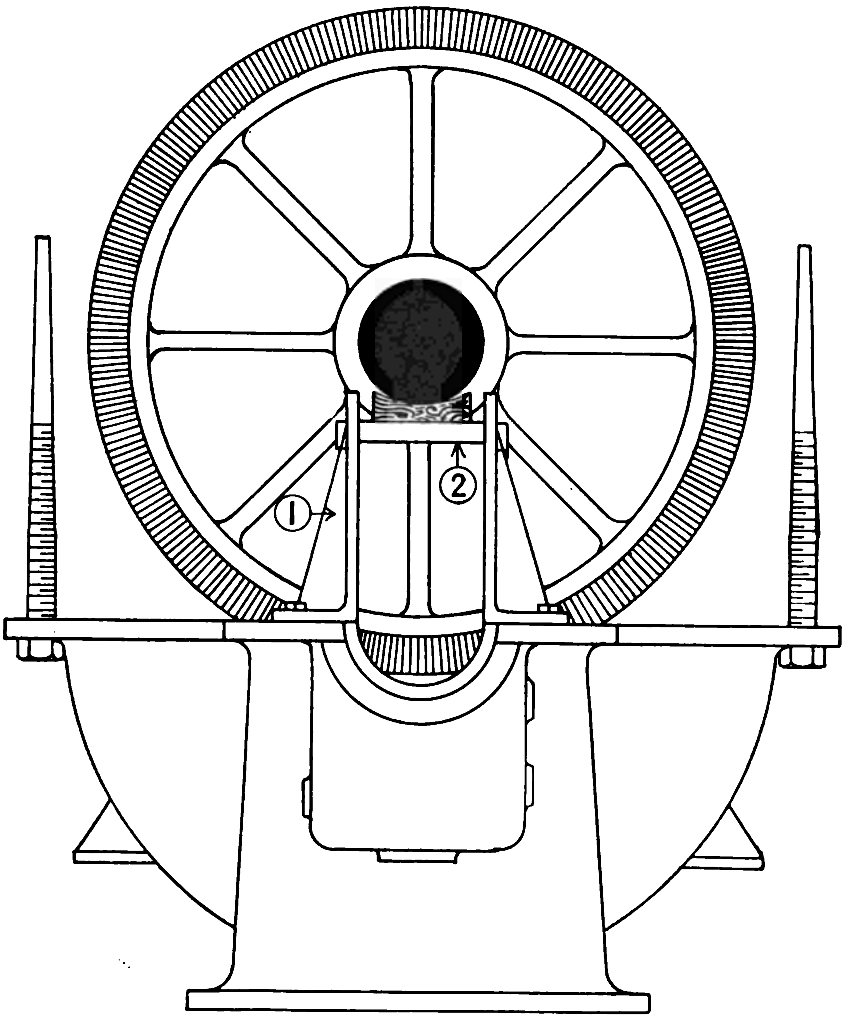


1. Strap for Lifting Rotor.

be about $\frac{1}{16}$ in. clear of blades in casing dummy. In the bottom of the casing between the rows of blades in each expansion melted wax is poured to a little more than what the blade clearance will be.

The rotor is revolved and cuts away any superfluous wax, leaving an amount equal to the clearance between the tip of the blades and the casing ; the blades are also gauged at the sides at the same time.

When the rotor is lifted the wax is taken out and gauged by



Rotor Clear of Casing and Resting on Carrying Bars.
1. Guide and Support Brackets. 2. "Carrying Bar" for Rotor.

micrometer. The bearings are now eased so as to ensure the requisite clearance on the sides, and wax is put on the rotor between the blades and also on the top half of the casing, which is put on, with the rotor in place. The rotor is revolved, and top half of casing lifted off and

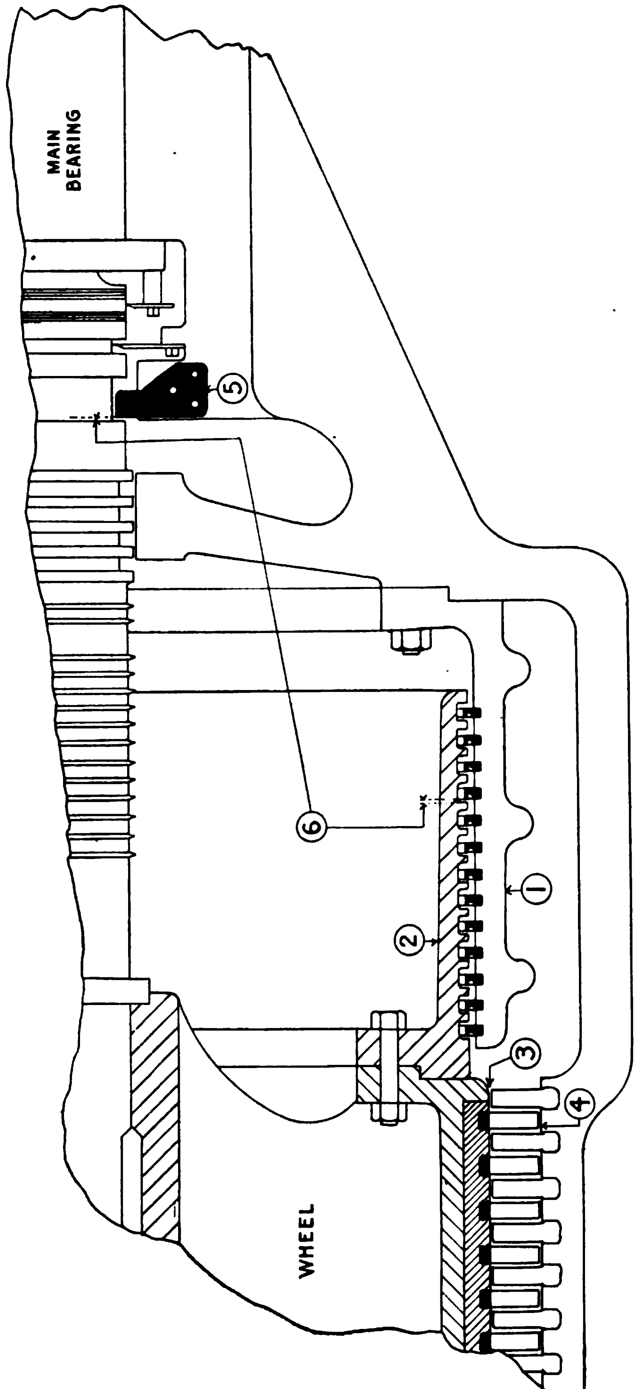
the wax gauged, which gives the top and bottom clearance of the rotor blades, and also the clearance of the casing blades. This having been adjusted to what is required, the dummy is now ground up. This is done by forcing the rotor up against the dummy blades while revolving, and ground emery and oil being poured in until all the blades on the dummy are at point of touch.

A gauge is now made showing the position of the rotor fore and aft. The rotor is put aft and the top half of the casing put on and bolted, and the dummy again brought into contact with the dummy blades and ground up on the top half until the rotor is at the same position as when the bottom half was ground, as ascertained by gauge. The thrust block is then adjusted and the longitudinal clearance of the rotor blades taken, also the radial and dummy clearance. The top main bearing bushes are leaded and the rotor lifted out, and casing finally cleaned and prepared for steaming to ascertain if the dynamic balance is correct. Guide columns are fitted for guiding casing and rotor into position, two at forward and two at aft end.

Steaming of Turbines.—In heating up turbines for steaming purposes, if the turbines are of large dimensions, it is usual to have heating steam pipes led into the casing at the forward and aft end. This ensures an equal heating up of the turbine.

In small or average size turbines there are no heating pipes fitted, the steam being admitted at the forward end and passing aft. There are different opinions regarding which is the best way to heat up, some authorities preferring to heat up the turbine as quickly as possible so as to ensure an equal expansion all over, and others to heat up gradually. Steam being admitted into casing, the rotor should be turned about every two or three minutes, as if this is neglected the steam entering on one side of the casing will heat up one side of the rotor excessively, causing it to expand on that side, and thus alter the alignment of the shaft and cause it to bind in the bearings. While heating, attention should be paid to the finger piece.

When the dummy is set to the required clearance the "finger piece" is also set to the same, having a clearance from the aft side of the groove on shaft equal to the clearance of the dummy. This should be gauged by feelers while heating up, more especially if for the first time, and should the dummy clearance decrease until the clearance is only .01, steam should be shut off so as to allow what heat has been imparted to transmit itself through the body of the wheels to the shaft. The shaft being expanded against the thrust will thus cause the rotor to expand in an aft direction. The above explanation is only applicable to steaming turbines in the shops, where no steam connections to the glands are fitted. The oil should be circulated through bearings for two or three hours before starting, and the filter opened and cloths cleaned so as to ensure that all grit and dirt has



Dummy and Finger Plate showing how Clearance is Measured.

1. Dummy Casing.
2. Dummy Piston.
3. Steam Admission to Blades.
4. 1st Expansion Blades.
5. Finger Plate.
6. Dummy Clearance and Finger Plate Clearance Equal.

been washed out of bearings. All drain connections should be left open, and it is preferable to have an ejector fitted and drains led away so that they can be left open while running, as owing to length of piping required to transmit steam to turbines, in most shops, there is bound to be a large amount of condensation. As before mentioned, when balancing, the turbines should be run up to 20 per cent. above their working speed to ensure a good dynamic balance.

When running a turbine in the shop under steam up to the required speed, after the steam is shut off the rotor usually continues revolving for some minutes before coming to a dead stop, thus proving the correctness of the balance.

In the case of large turbine rotors after steaming the drums have been found slightly out of truth due to unequal expansion; this necessitated further turning up in the machines, and in some cases filing up of some of the blade rows at certain positions to maintain the required tip clearance.

It has also been discovered that the blade tip side clearance is affected by the weight of the upper half casing, as in one case tested the side clearance measured $\frac{15}{1000}$ in. with the upper half cover off and $\frac{60}{1000}$ in. with the cover bolted down. This is understood to be due to the weight of the upper casing producing a "spring" on the lower half and thus opening up the cylinder diameter horizontally, with resulting increase of blade tip side clearance.

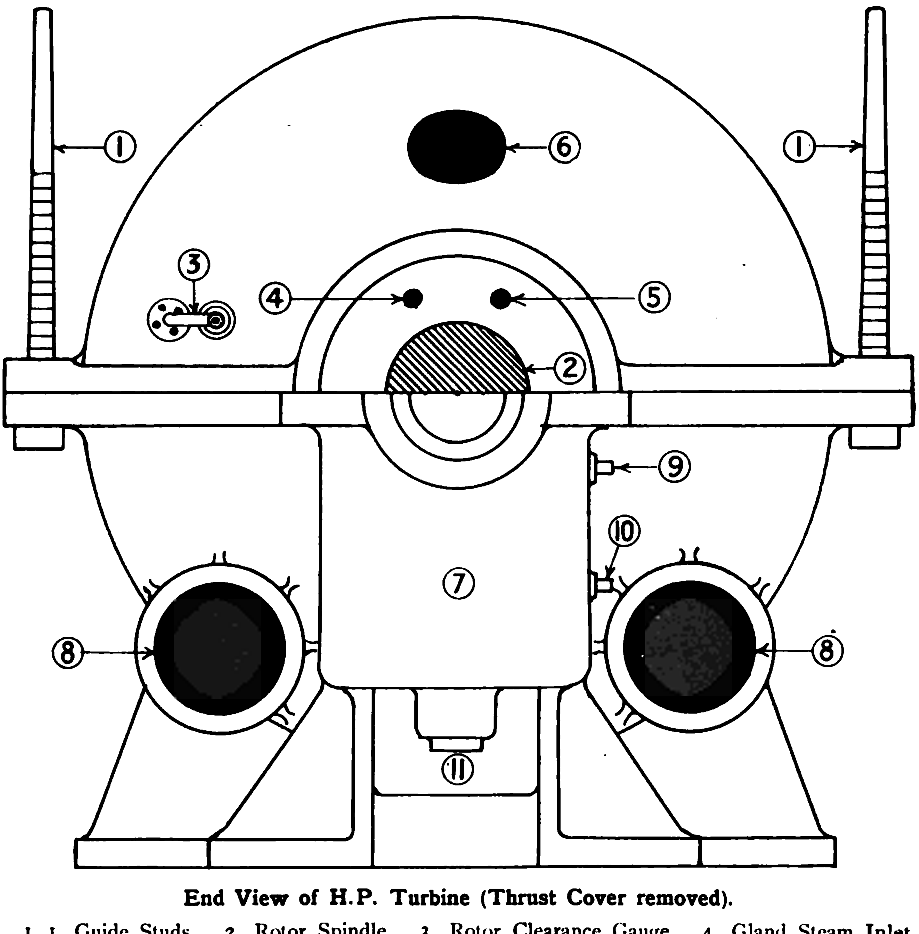
Guide Columns.—Four long guide columns are screwed into the lower half turbine casing, and these are intended to maintain the upper half casing in its correct position when being raised or lowered, and thus prevent possible damage to the blades by contact. These columns are placed two at each end, and they are carefully marked in inches, and parts of inches, to allow of the cover being raised or lowered evenly throughout the length of the turbine, and thus obviate canting.

Turbine Casing Drains.—A drain pipe and cock or screwdown valve connects the dummy casing at the forward end of each turbine casing with one of the last expansions aft. This allows any condensed water to drain out from the higher forward position of the casing to the lower after position. These drain connections are only kept open when the turbines are stopped temporarily.

L.P. Turbines.—Large double drain pipes and specially light non-return valves are fitted at the after end of each L.P. turbine, and these are connected to the "wet" or ordinary air pumps which are kept running while the turbines are stopped temporarily. This ensures the withdrawal of any condensed water from the turbine casings. To prevent air finding its way back into the L.P. turbine casings, a **U** bend is arranged on the air pump suction pipe to act as a "water seal," and as this bend contains water, the admission of air is prevented even should the air pumps cease working. The non-return

valve mentioned above would prevent the possible return of water back from the pump into the turbine casings.

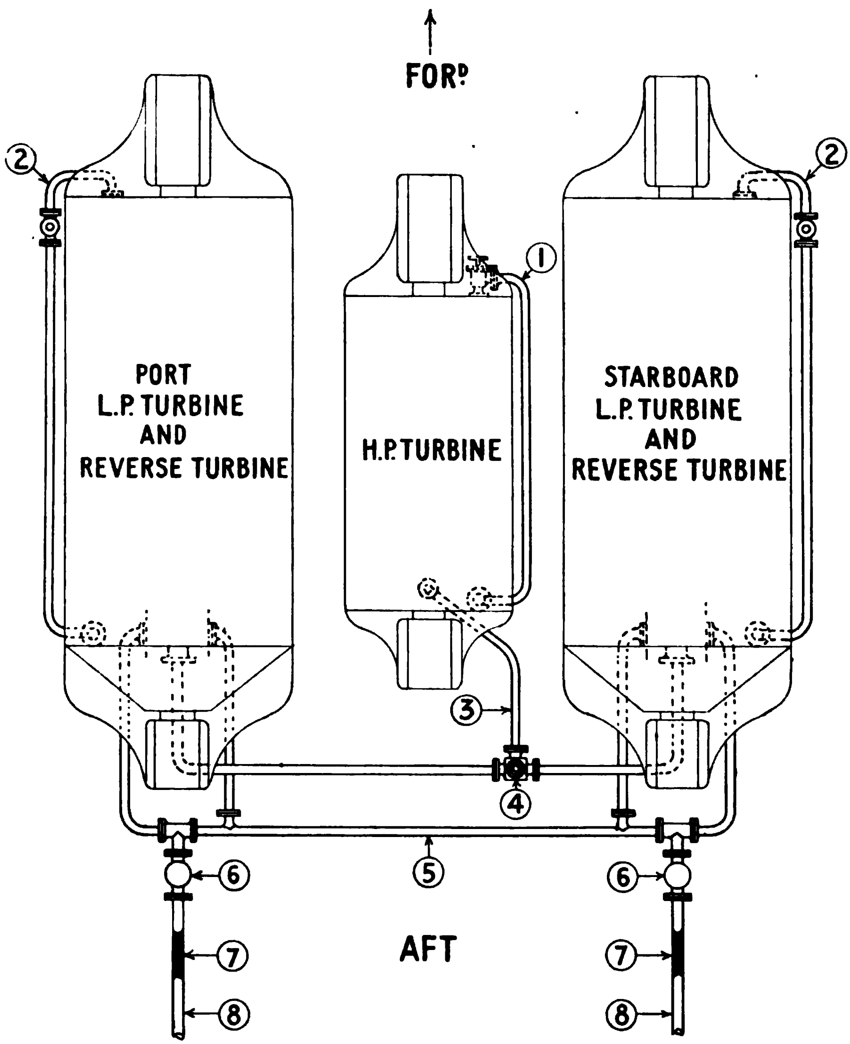
H.P. Turbine.—The drain from the after end of the H.P. turbine is merely led to the drain pipe connection of either of the L.P. turbines, a cross or "change over" connection to either L.P. being arranged by means of a two-way cock.



End View of H.P. Turbine (Thrust Cover removed).

1, 1. Guide Studs. 2. Rotor Spindle. 3. Rotor Clearance Gauge. 4. Gland Steam Inlet. 5. Gland Steam "Leak-off." 6. Hand Hole. 7. Thrust Seating. 8, 8. Steam Inlets to Turbine. 9. Oil Supply. 10. Cooling Water Inlet. 11. Oil Outlet.

NOTE.—The water which is constantly drained off from the L.P. turbines when running is formed by the adiabatic expansion of the steam in the turbines. The H.P. exhaust pipes to each L.P. turbine are placed low down to allow the water condensed in the H.P. turbine to pass easily to the L.P. turbines, and afterwards be drained off by the connections aft to the "wet" air pumps. (See sketch facing page 120.)



Turbine Drain Connections.

1. Drain from forward to after end of H.P. Turbine.
2. Drain from forward to after end of L.P. Turbines.
3. Drain from after end of H.P. Turbine to after end of either L.P. Turbine.
4. Two-way Cock for changing over.
5. Drain from after end of both L.P. Turbines to "Wet" Air Pump.
6. Light Non-return Valves.
7. U Bend in Pipe to act as "Water Seal" and prevent return of Air.
8. "Wet" Air Pump Suction Pipes.

NOTE.—The connections marked 1, 2, and 3 are only open when the turbines are stopped, and the connections marked 5, 6, 7, and 8 are open when the turbines are either running or stopped (temporarily).

All casing drains can be opened up to the bilges when required, and this is done when the engines are rung off, so as to get rid of the vacuum existing in the turbine casings.

The two drains led from the H.P. exhaust pipes to the L.P. casings aft are often just kept eased off the face when running, so as to allow the condensed water only to be drained off, while in other cases these are shut altogether when running.

Rotor Screw.—When the rotor requires to be moved longitudinally, a pin is screwed into the forward end of the spindle, which is tapped out for the purpose, and by means of a dog arrangement and double nuts the rotor can be moved forward or aft as required. It is sometimes necessary to disconnect the first length of shafting when screwing up the rotor as described.

Steam Connections.

The usual steam connections on the rotor casing are as follows:—

Ahead Turbines.

1. **Steam Inlet at Forward End.**—In the H.P. turbine this is direct from the boilers, and in the L.P. turbines from the H.P. exhaust.

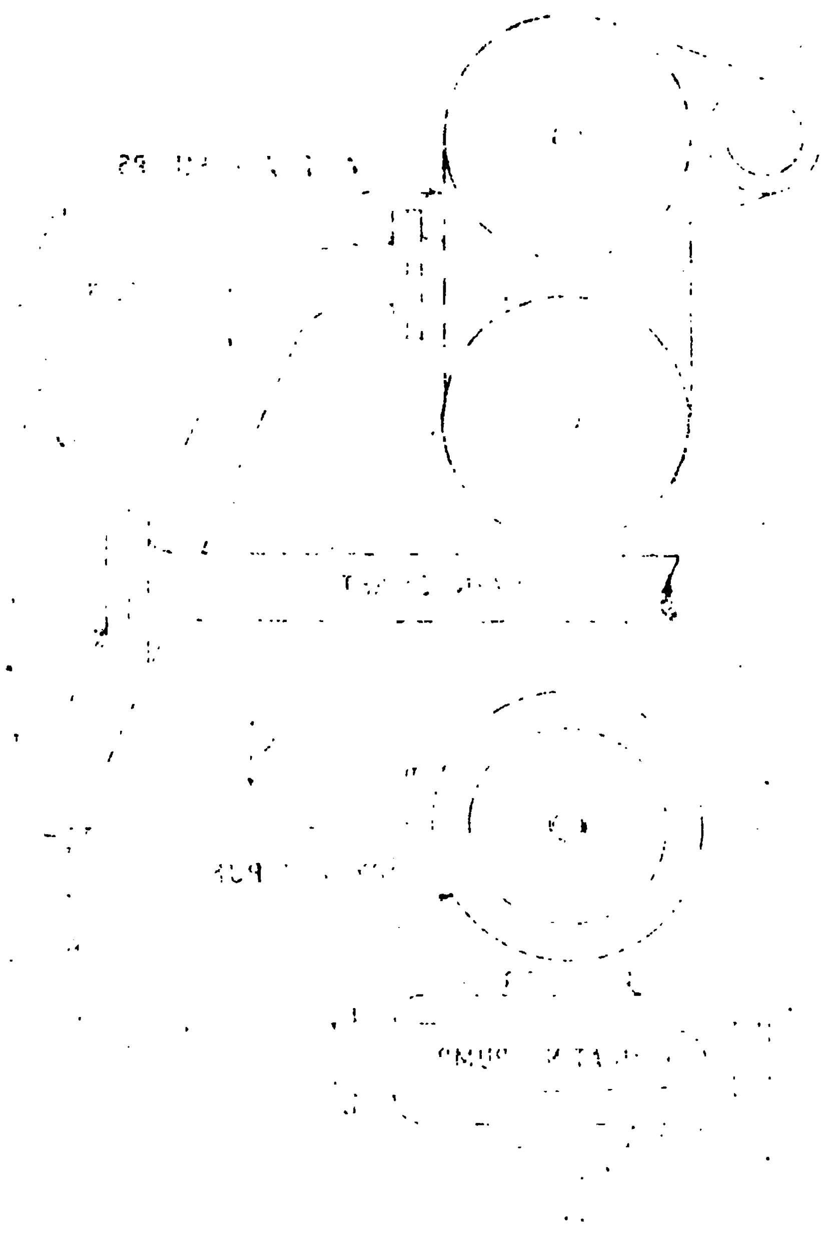
2. **Exhaust at After End.**—In the H.P. turbine this leads to the two L.P. turbines as initial steam, and in the L.P. turbines direct to the condenser.

3. **L.P. Turbine Non-return Valves.**—These valves, arranged to prevent the return of steam from the L.P. to the H.P. turbines, are fitted in a chest on the L.P. casing where the H.P. exhaust branch connects, and being loaded by means of two springs, only allow of the admission of steam from H.P. to L.P. casings, but close immediately steam attempts to pass back to the H.P. from the L.P. These valves come into action when high-pressure steam is supplied direct to the ahead L.P. turbines in manœuvring or working with the outside shafts only.

4. **Direct Steam to L.P. Turbine.**—This supplies high-pressure steam direct to the ahead L.P. turbines, and is required when running ahead with the outside shafts only.

5. **Bye-Pass Steam.**—This consists of either one or two hand-valves and pipe connections leading from the 1st to the 3rd expansion of the H.P. turbine to admit high-pressure steam direct to the 3rd expansion if required to increase the power, or to create an equalising pressure in starting up and prevent possible damage to the blades by excessive vibration.

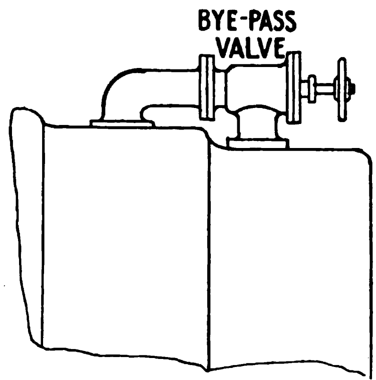
6. **Escape or Relief Valves.**—Escape valves, loaded to a suitable pressure, are fitted usually on the top of the two L.P. turbine casings forward and aft, and one on the H.P. turbine casing aft.



7. **H.P. Gland Steam.**—Steam of reduced pressure is admitted to the H.P. gland pockets only when heating up to allow of gradual expansion of the metal; but when running this connection forms a "leak-off" to one of the L.P. turbines; this is arranged for by means of a two-way cock.

8. **L.P. Gland Steam.** — Both ends of each L.P. turbine have low-pressure steam admitted to the gland pocket to prevent the admission of air. No "leak-off" is fitted, and the pressure of steam varies from 1 to 4 lbs.

In both the H.P. and L.P. turbines the exhaust steam from the last ring of blades flows into the *inside* of the drums, and the steam glands are therefore required to pack the spindle against outflow of steam in the case of the H.P. turbine, and against the admission of air in the case of the L.P. and reverse turbines. The pressure inside of the H.P. rotor drum is usually somewhat between 10 and 25 lbs., and



Bye-pass from 1st to 3rd Expansion (on H.P. Turbine only).

the vacuum inside the L.P. rotor drum approximates to that carried in the condenser, being probably between 2 or 3 lbs. less. As before stated, the H.P., L.P., and reverse dummies act as packing to prevent the outflow of the higher or admission pressure steam from the turbine casing into the atmosphere; whereas the steam glands on the rotor spindles are only required to pack the turbine against the exhaust or lower pressure steam.

9. **Dummy "Leak-off."**—When a dummy leak-off is fitted, two pipes are usually led from the ahead dummy casings to the 3rd expansion of the turbine, where the pressure is less, to take away any steam which may leak through the first series of dummy rings and grooves.

Reverse Turbine.

10. **Direct Steam.**—The reverse turbines only require one connection, that of the direct steam at the after end, which, after expand-

ing through the reverse turbine blades forwards, exhausts into the condenser by the common exhaust branch. Small drain holes are fitted in the bottom of the reverse turbine casing.

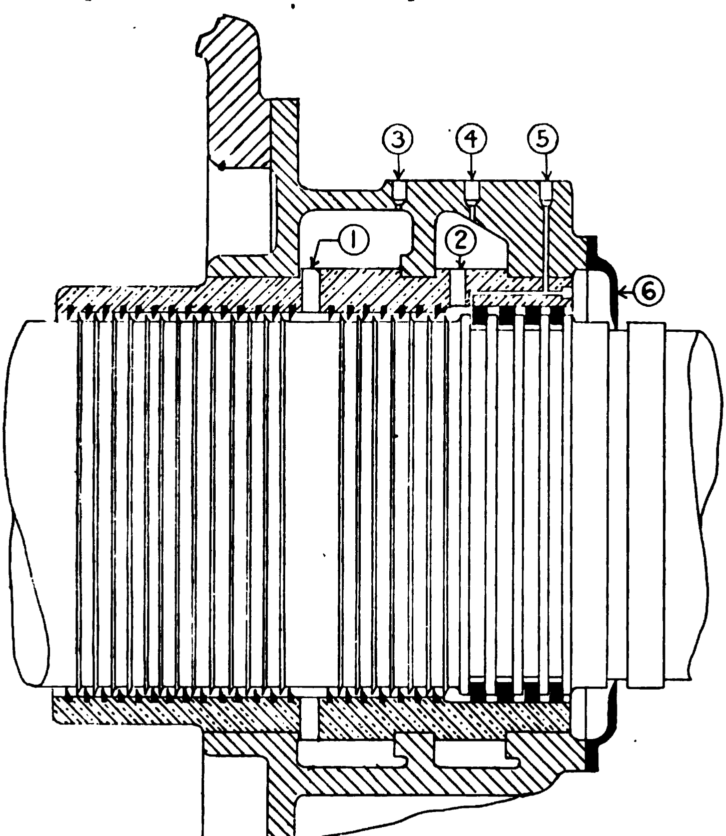
Steam and Reverse Valves.—Steam can be admitted by the large hand-controlled valve to the H.P. turbine direct, and so through the series of turbines, so that all are working at once. Two smaller hand-valves also admit full-pressure steam to a pair of chests, each containing a piston valve which is actuated by patent steam and hydraulic gear. Round ports are cast in the chest, top and bottom, one end admitting steam to the ahead L.P. turbine, and the other admitting steam to the reverse L.P. turbine, and the valve is moved up and down by the gear to uncover the ports and admit steam from the centre of the piston valve as required, the steam at the same time being admitted to the chest itself by means of the large hand-valve referred to.

This constitutes the reversing and manœuvring gear.

Steam Glands.

The steam glands on turbines differ greatly from the steam glands on the reciprocating engine. In small turbines the usual type of gland consists of a series of brass ramsbottom or split rings fitted into grooves in the shaft, over which a sleeve of hard grained cast iron is fitted. The rings are usually of phosphor bronze. They are turned to a larger diameter than the inside of the gland, then so much is cut out, so as to give an opening at the joint of $\frac{1}{32}$ in. to $\frac{1}{16}$ in. when rings are in place inside of sleeve. This opening varies according to size of ring and the expansion of same when heated. Another style of gland, and one being largely adopted, is a combination of the ramsbottom ring and brass fin blades similar to those fitted in the radial or astern dummy. This form is usually termed labyrinth packing. The brass blades are fitted into grooves in the rotor spindles and caulked into place; they are then knife edged, and have a radial clearance at the tip of .010 in. to .020 in. The gland or sleeve enclosing the same is in two halves, and is of hard grained cast iron, grooved and bladed similar to the spindle. The blades in the spindle fit in between the blades in the sleeve. On the outer end of the shaft there are usually four brass ramsbottom rings which bear against another sleeve which is bolted to end of the casing (see sketch). Between the labyrinth packing and the ramsbottom rings a space is left which forms a pocket or receiver, and from this point the gland escape or leak-off takes place. This in an H.P. turbine is led by piping to the 2nd or 3rd expansion of the low-pressure turbine. The principle of the labyrinth packing is the same as in the radial dummy. The steam is wire-drawn in passing the tip of the blades, and then expanded into the space between the blades, and so on until it reaches the escape or leak-off pocket. Any leak-off that takes place past the four ramsbottom rings is also collected, and led away by a vapour

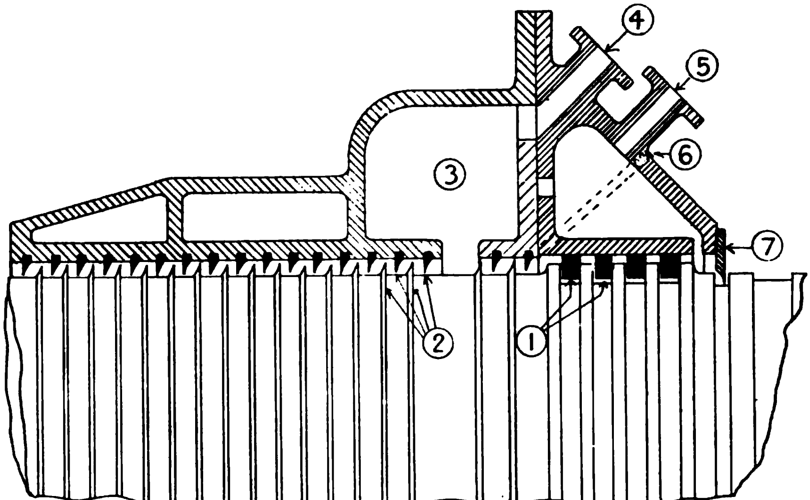
pipe either to the atmosphere or to the hot-well tank. In the case of a low-pressure gland, where the interior of the turbine is in a vacuum, the admission of air must be prevented: to do this, steam is admitted into the pocket between the ramsbottom rings and the labyrinth packing at a pressure from $1\frac{1}{2}$ to 3 lbs. per sq. in., which low pressure is sufficient to prevent the admission of air



Steam Gland (Naval Type).

1. Leak-off Ports (Inner).
2. Leak-off Ports (Outer).
3. Leak-off Pipe Connection.
4. Leak-off Pipe Connection.
5. Oil to Gland Rings.
6. Oil Baffle Plate.

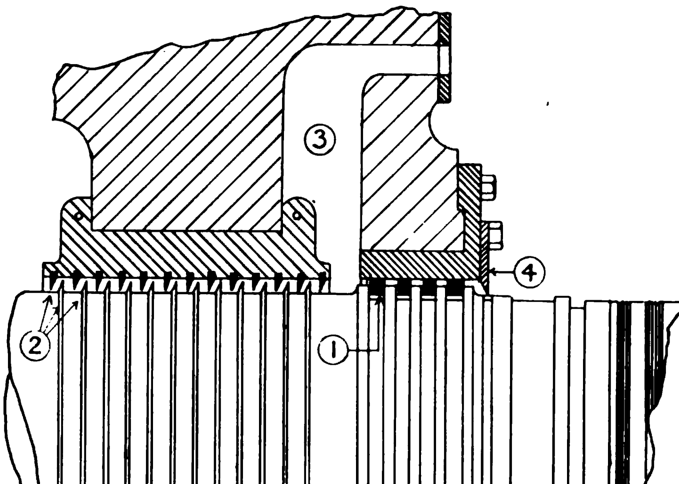
which would otherwise affect the vacuum. On the high-pressure gland there is also a steam inlet connection fitted to the leak-off pocket, and this connection is used when heating up the turbine preparatory to running: this ensures the expansion of the shaft in relation to the casing and rotor. By opening steam on the forward end and heating up the shaft, the shaft will expand and press against thrust rings, and so force rotor aft, and increase or keep the dummy clearance at the



Steam Gland.

(For large-sized Turbines.)

1. Ramsbottom Rings.
2. Radial Rings.
3. "Leak-off" Pocket.
4. Pipe Connection.
5. Vapour Escape.
6. Oil to Gland Rings.
7. Baffle Plate.

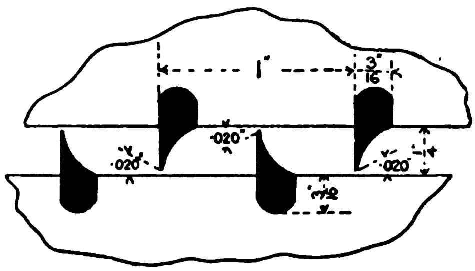


Steam Gland.

(For average size Turbines.)

1. Ramsbottom Rings.
2. Radial Fin Rings.
3. "Leak-off" Pocket.
4. Baffle Plate.

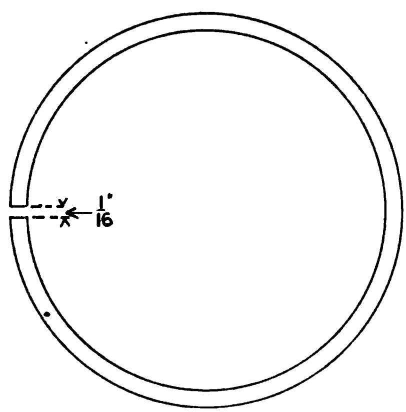
original setting. Should the shaft be left cold, and steam admitted into the turbine, the rotor will begin to expand and decrease the dummy clearance, but by warming the shaft at the same time this clearance can be maintained and the expansion of rotor neutralised. Shutting off steam to forward gland and opening steam to aft gland will send rotor forward and *vice versa.*



Gland Radial Rings.

(Corners rounded off to maintain strength of Turbine Spindle and of Case.)
From 11 to 14 rows of these are fitted to each Gland.

An impermeator is usually fitted for supplying oil to gland rings. Pressure gauges are fitted connecting to leak-off and vapour pockets, and drains are led from bottom of glands usually into hot-well tank, but in some cases only into bilges.



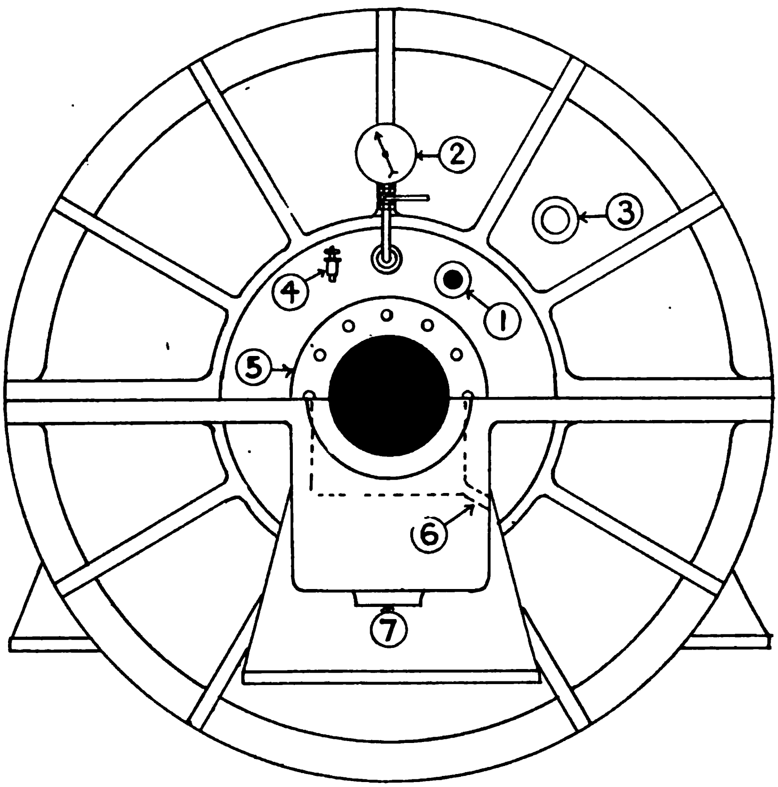
Ramsbottom Rings of Gland.

(Phosphor Bronze.)

Turbine Mountings and Connections.

A high-pressure turbine has pressure gauges fitted showing initial and terminal pressure. An escape valve is fitted at forward end in the steam receiver space, and a drain is also led from this part into the exhaust pipe to the low-pressure turbine. This is termed an annulus

drain. On the steam glands at the leak-off and vapour pockets steam valves are fitted, also a connection admitting steam to the interior of the glands. If the exhaust pipe connection on the H.P. turbine is from the bottom half of the casing, a drain connection from each pipe is led to the bottom of after end of low-pressure casing, with a shut-down valve between same. An extended spindle is fitted

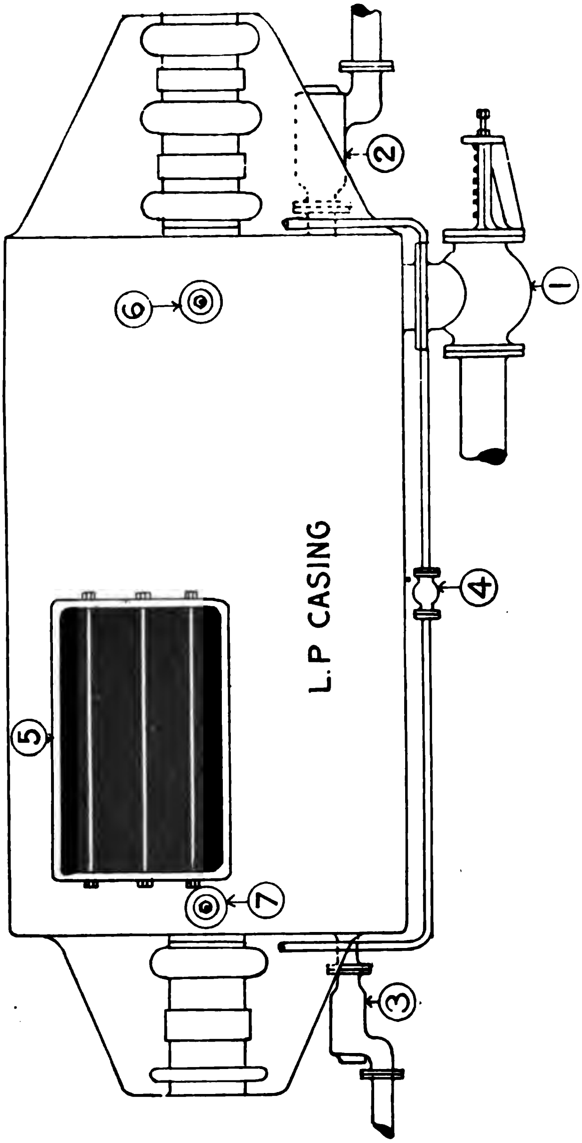


End View of H.P. Turbine.

1. Gland "Leak-off."
2. Pressure Gauge for Gland.
3. Micrometer Gauge.
4. Oil to Glands.
5. Gland Case.
6. Drain from Gland Pocket to Hot-well.
7. Oil Drain to Cooling Tank.

to these valves, and led up above the platforms so as to be easily worked. The after end of low-pressure turbines being in a vacuum, water which accumulates in the high-pressure turbine is drawn by the drain pipes into the L.P. casing. A connection is also made from bottom of low-pressure after end to air pump suction, so that in heating up turbines water can be drawn from both H.P. and L.P.'s to air pump and then into hot-well.

Starting and Reverse Steam Valves.—The steam in its passage from the boiler to the turbines passes through the boiler stop-valves, then through a master valve; next past the throttle or governor valve,

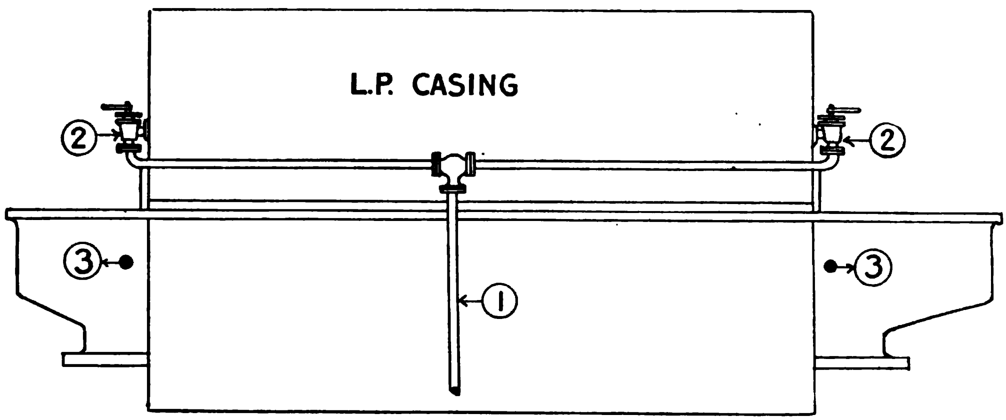


Plan of L.P. and Reverse Casing.

1. Steam Inlet to L.P. from H.P. Exhaust.
2. Ahead Steam direct from Boilers.
3. Astern Steam direct from Boilers.
4. Steam to Glands.
5. Exhaust to Condenser.
6. Relief Valve on Forward End.
7. Relief Valve on After End.

then the main stop-valve, from which the pipes to the high-pressure turbine are connected. After passing through the H.P. turbine the steam is led through two pipes, one to each L.P., and then it passes

through the exhaust pipe into condensers. When manœuvring, that is, working the two outside shafts, steam is admitted to these manœuvring valves from a connection below the throttle valve into two chests, one for port and one for starboard. In these chests are two valves, one admitting steam to low-pressure turbines and the other steam to astern turbines. Governors are fitted sometimes, of the ball type, connected up to the throttle valve, this valve being placed between the boilers and the main stop-valve and manœuvring valves, should turbines race; then if throttle valve shuts, steam is cut off completely. In large turbines having the astern turbine separate from the L.P., a steam heating connection is fitted so as to keep this turbine warmed up, ready to start when required.



L.P. Casing Connections.

1. Steam to Glands. 2. Cock to admit Steam to Pocket of Gland. 3. Gland Drain to Hot-well.

Dummy Adjusting Gear.—For turbines of large power and weight, such as those fitted in battleships, cruisers, and large passenger steamers, the thrust block adjusting is usually arranged as shown in the sketches. After the lower and upper half blocks are set by the worm gear shown, to the required dummy clearance, the two are locked by means of wedge keys (one for each half block) which are screwed up by the nuts fitted on each end of the keys for that purpose. Indicators are fitted which register the amount of longitudinal movement given to either the lower (ahead) or upper (astern) half of the thrust block, which, of course, regulates the dummy clearance. Movement is given to each half block by a pair of horizontal shafts which are keyed to prevent rotation. On these shafts the worm wheels are screwed; when, therefore, the worm is revolved, the worm wheel of the shaft also revolves, and as this wheel is prevented from moving

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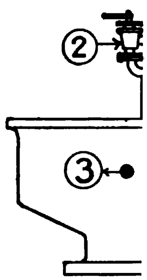

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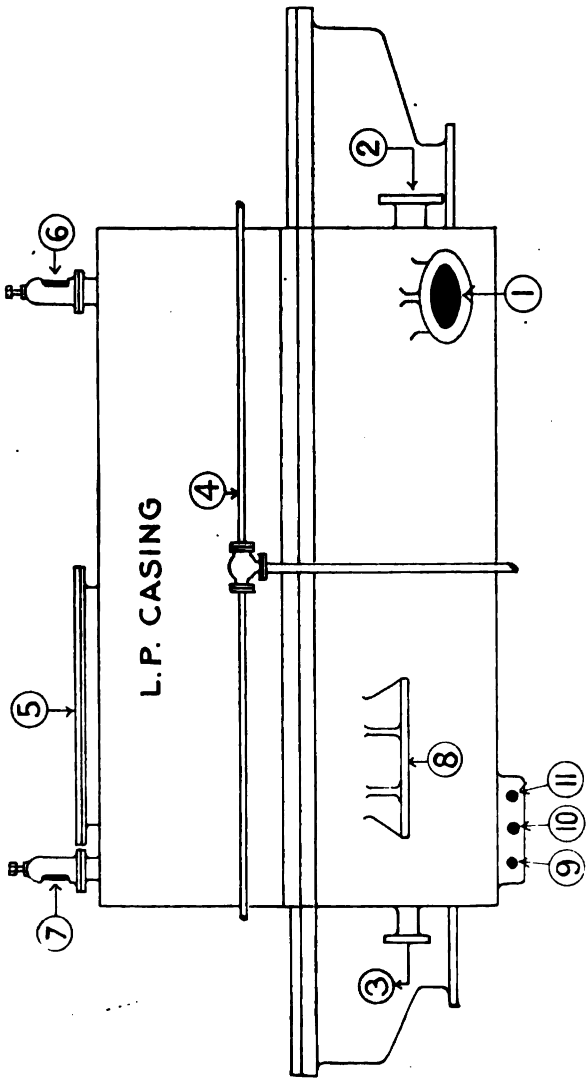
longitudinally the shafts move instead, either to right or left, and so change the longitudinal position of the half blocks as required. Before either half block can be moved the wedge key must be

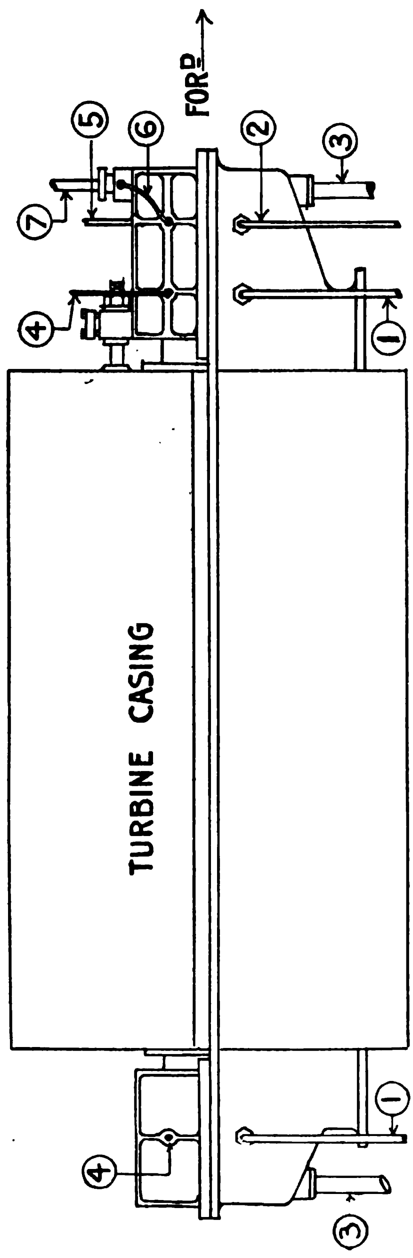


L.P. Turbine Casing Connections.

1. Admission Steam from H.P. Exhaust.
2. L.P. ahead Steam direct from Boilers.
3. L.P. astern Steam direct from Boilers.
4. Reduced Steam to Glands.
5. L.P. exhaust to Condenser.
6. Escape Valve at admission end.
7. Escape Valve at exhaust end.
8. Feet for bolting down.
9. Drain to "Wet" Air Pump.
10. Drain from H.P. to L.P.
11. Connection to Bilges.

slackened back, and after the dummy adjustment is made the wedge key requires to be screwed up hard to lock the block in position. The taper of the wedge is $\frac{7}{16}$ in. per foot.

Instructions for Dummy Adjustment.—When the dummy



Turbine Connections.

1. Oil Inlet to Main Bearing.
2. Oil Inlet to Thrust.
3. Oil Drain to Cooling Tank.
4. Oil Pressure Gauge Connection for Main Bearing.
5. Oil Pressure Gauge Connection for Thrust.
6. Oil to Tachometer Gear.
7. Spindle of Tachometer Gear.

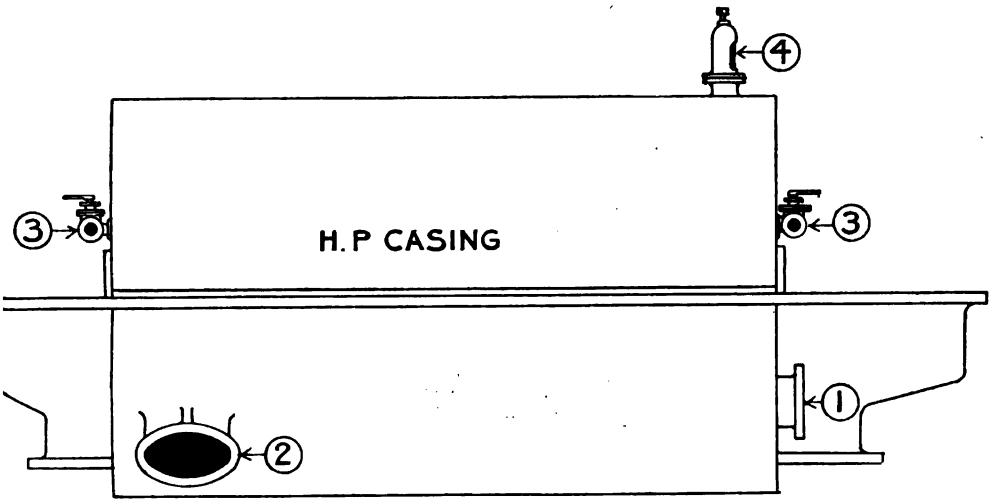
adjustment is obtained by means of the taper keys and worm gear described above, the instructions as to setting are as follows:—

(1) When the dummies are in contact and the top and bottom thrust collars hard up against thrust rings, the indicators read zero when nuts are tightened up.

(2) Each revolution of indicator nut is equal to .003 in. forward or aft movement of block.

(3) Before moving the top half of block aft, or the bottom half forward, the top or bottom wedges must be slackened away.

(4) When running, the top wedge indicator should read .010 more than bottom one to allow of oil clearance film.



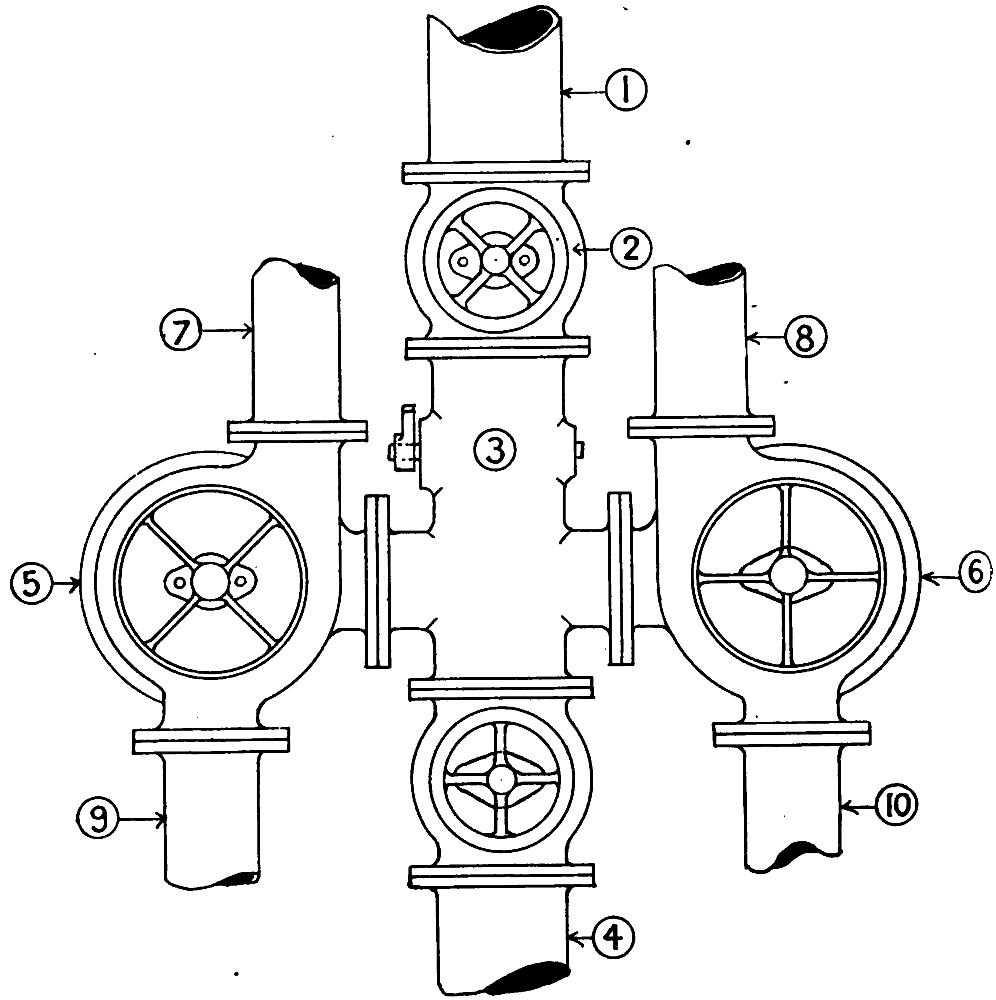
H.P. Casing and Connections.

1. Boiler Steam Inlet.
2. Exhaust Steam Outlet to L.P. Turbines.
3. Reduced Steam to Glands.
4. Relief Valve.

To Reduce Dummy Clearance.—Ease back bottom half block aft, slack indicator nut until it reads the required clearance, then screw up back nut to secure wedge. With bottom worm gear screw bottom half block forward till it bears on wedge, then with top gear screw top half forward, following up with top wedge by easing away indicator nut and tightening up back nut until the top indicator gives the same reading as the bottom one, *plus .010*; finally screw top half of block aft until it bears hard on wedge.

To Increase Dummy Clearance.—Screw bottom half of block aft and follow up with wedge by easing back nut and screwing up indicator nut until the required clearance is obtained, now screw up back nut to secure wedge, and screw bottom half block forward until it bears hard on wedge. It should be observed that the thin end of the bottom wedge and the thick end of the top wedge are on the same

side as the indicators; also, that in screwing the rotor either forward or aft it will require to be turned meanwhile.



Ahead and Reverse Steam Valves to Turbines.

(On starting platform.)

1. Steam from Boilers.
2. Master Valve.
3. Throttle Valve.
4. Steam to H.P. Turbine.
5. Ahead Steam to Port L.P. Turbine.
6. Ahead Steam to Starboard L.P. Turbine.
7. Steam to Port Astern Turbine.
8. Steam to Starboard Astern Turbine.

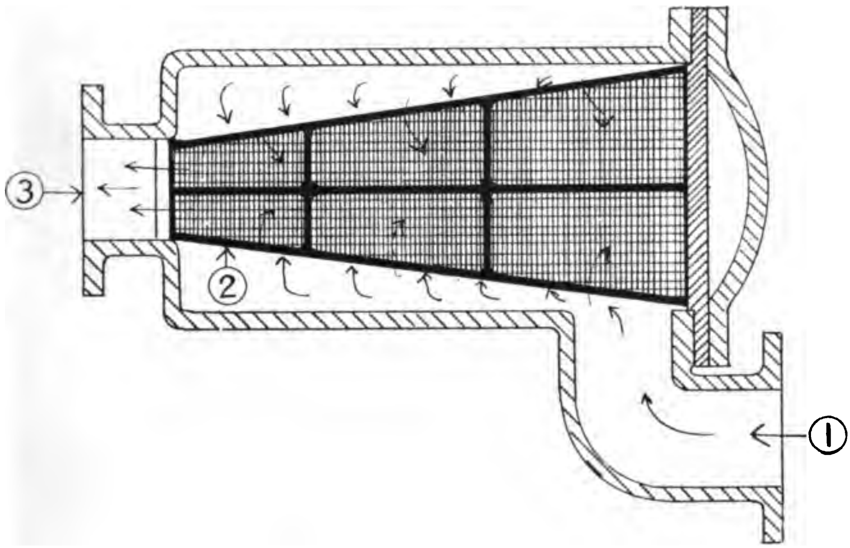
Strainers.—On each of the steam inlet flanges on the turbines strainers are fitted, through which the steam passes on its way to the

ERRATA.

The index for the sketch on page 132 should read as follows:—

1. Steam from Boilers.
2. Master Valve.
3. Throttle Valve.
4. Steam to H.P. Turbine.
5. Manœuvring Valve for Port L.P. Turbine.
6. Manœuvring Valve for Starboard L.P. Turbine.
7. Steam to Port Astern Turbine.
8. Steam to Starboard Astern Turbine.
9. Steam to Port L.P. Ahead Turbine.
10. Steam to Starboard L.P. Ahead Turbine.

turbines, and any solid matter contained in the steam is trapped and collected in the strainer casing; their purpose is to prevent any solid matter from passing into the interior of the turbine. The strainer cartridge, as it is called, consists of a framework covered with brass perforated plates or of brass grids (see sketch).

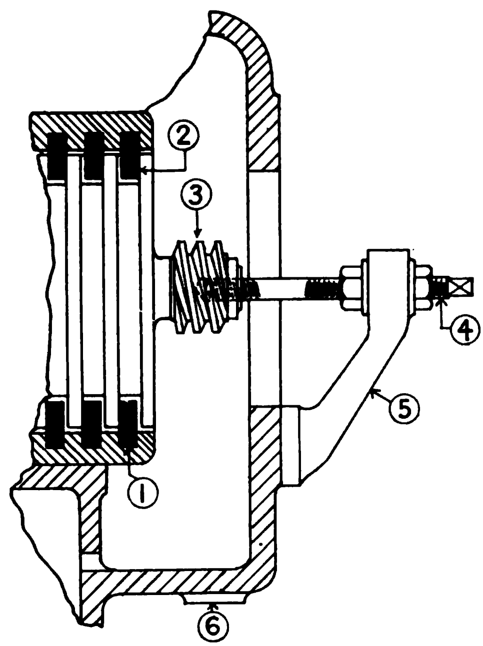


Steam Strainer.

1. Steam Inlet. 2. Brass Wire Mantle or Cartridge. 3. Steam Outlet to Turbine.

Calibration of Turbine Shafts.—To calculate the shaft horsepower by torsion meter it is requisite to obtain a constant for the torque of the shaft as this varies with different makes and size of shafting. To attain this result the shafting requires to be calibrated, and the usual method of doing this is as follows:—The shafting between the turbine and tail shaft is laid along the shop floor and bolted together by the coupling bolts belonging to the shaft. This length of shafting is levelled up, and is laid on wood blocks faced with sheet iron. The forward end of the shaft is rigidly bolted to a lever or arm, and on the end of same large weights are laid. At the other or after end of the shaft another lever is bolted, care being taken to see that in both cases there is no play in the holes. This lever is adjusted so that it travels an equal distance above and below the centre of the shaft when the weights are applied. At equal distance along the shaft light clamps are fixed, which have extended arms, on the end of which is fixed a piece of planed wood. The distance from centre of shaft to the wood varies according to size of shaft. These arms are also levelled, and in front of them suitable platforms are arranged on which to work a surface gauge. The movable end of the shaft is

left unweighted, and lines are drawn across the wood on the end of the arms. Weights are now put on the lever at aft end of shaft, these weights being applied at a given distance from centre of shaft, and when sufficient weight has been applied other lines are drawn across the wood on the end of arms. More weight is applied and more lines drawn as the weight is increased. The distances between the marks are measured and indicate the torque or twist of the shaft due to the weight applied at the leverage selected. From this operation constants are obtained which are used in the formula employed in calculating the shaft horse-power by torsion meter readings.



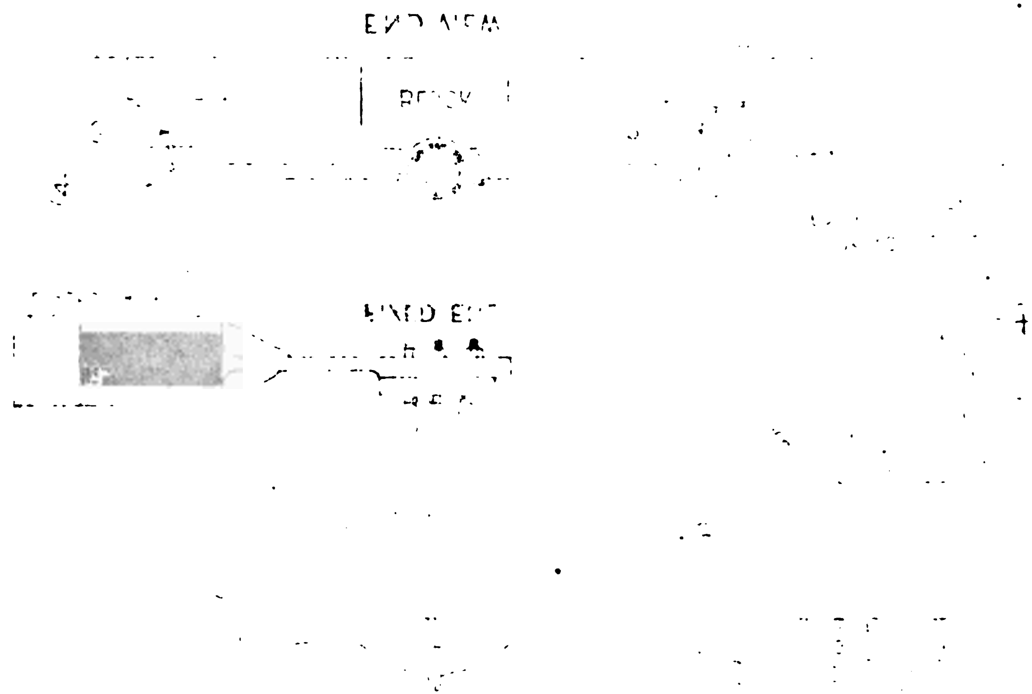
"Screwing-up" Bolt.

1. Ahead Thrust Rings. 2. Astern Thrust Rings. 3. Worm for Counter Gear. 4. Screwing-up Bolt. 5. Bracket. 6. Oil Drain.

Forced Lubrication in Turbines.

There are several systems of forced lubrication. The simplest arrangement is to have the pump drawing direct from a supply tank, through a filter, and discharging direct into the bearings, and then draining back to supply tank. In another system, the pump draws from the supply tank and discharges through a filter, which consists of a vessel having inside two brass perforated plates, with a fine gauze screen between: the oil passes from the filter to an overhead supply tank, placed high up in the engine-room, and on the top of the tank

FREE END



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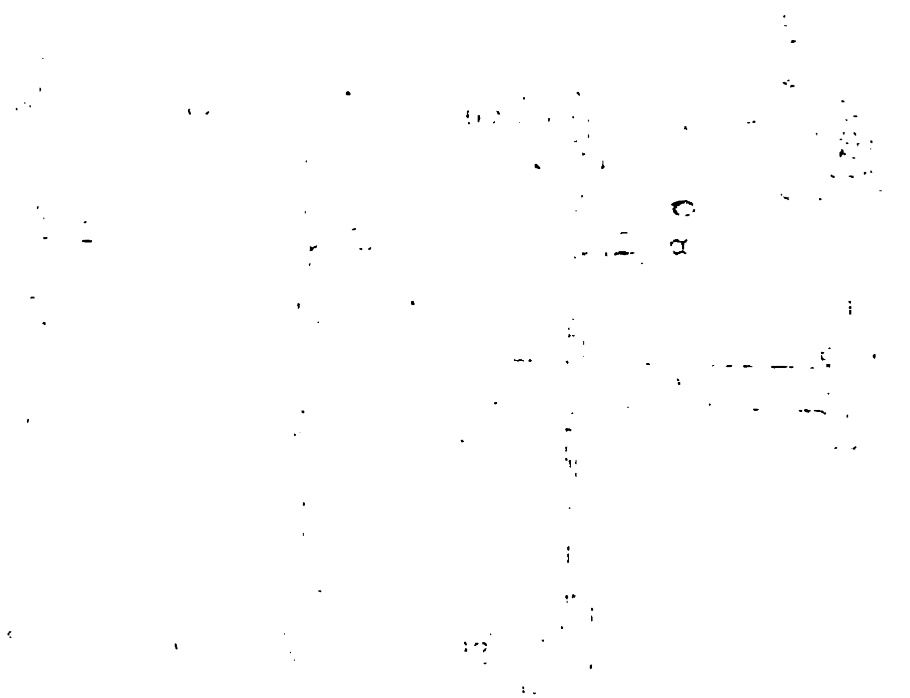

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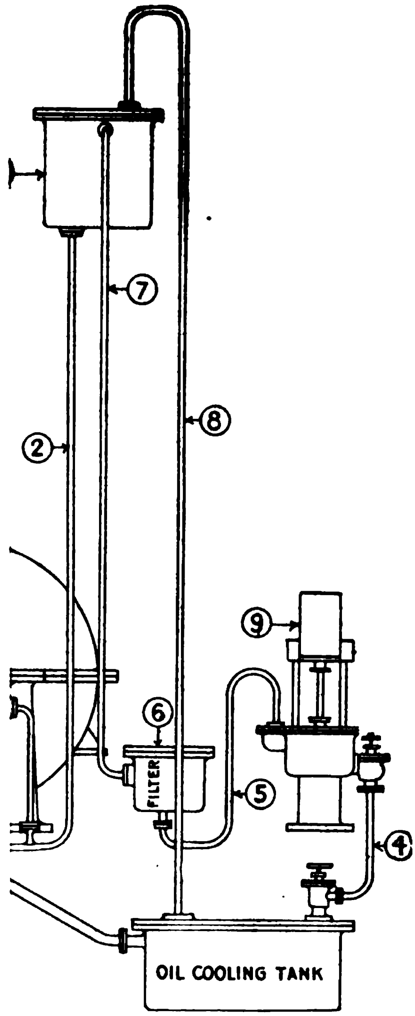

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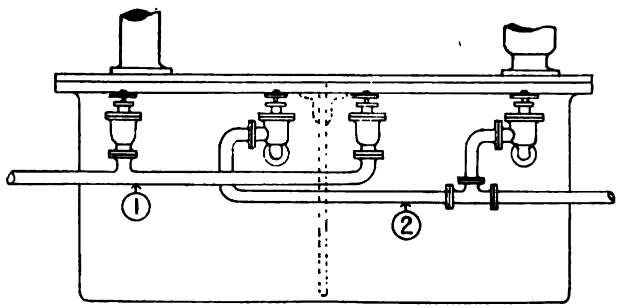
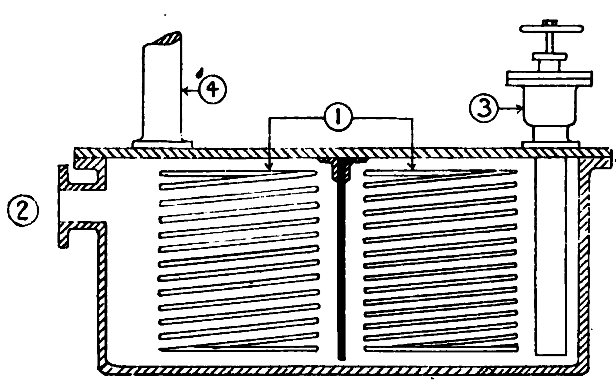
is an overflow pipe, which is led back to the suction tank down below: the oil flows from the overhead tank to the bearings by gravitation, and is then led back by drains to the suction tank below. It is preferable to have the supply tank on top of sufficient size so as to ensure that, in



Oil Cooling Tank.

1. Circulating Water Inlet to Coils. 2. Circulating Water Discharge from Coils.

the event of the pump breaking down, a sufficient supply of oil will be led to turbine bearings, until turbine is stopped or reserve pump started. The suction tank is usually fitted with cooling coils, through which water passes from a connection on the main circulating inlet. This en-



Oil Cooling Tank (Section).

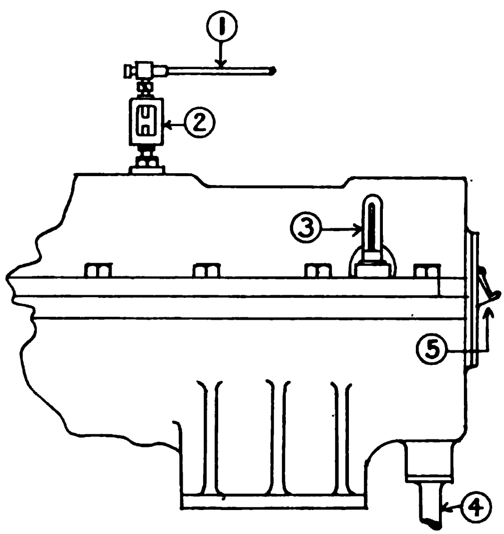
1. Cooling Water Coils.
2. Oil Inlet from Bearings.
3. Suction Valve to Pump.
4. Overflow from Supply Tank Overhead.

sures the oil being kept at as low a temperature as possible, as, owing to the heat imparted to the shaft of the turbine, the oil is soon heated up. On the main bearing covers test cocks are fitted so as to allow of inspection to see that oil is circulating, and hand access doors are arranged at the forward and aft end of the cover to allow of the bush

being felt by hand. Sight drains are sometimes fitted, consisting of a lantern casing having glass inserted in the sides, through which the oil may be observed as it drains back to tank. Gauges are also fitted to the bearings to indicate the pressure on same. This is usually from 4 to 10 lbs. per sq. in. In large bearings thermometers are fitted, showing the temperature of the oil which passes through the bearings.

It is of the utmost importance that the oil service be absolutely regular and uninterrupted, as if not, complete disablement of the turbine may result.

The regular maintenance of the oil service is of the highest importance for the successful running of the turbines, as should



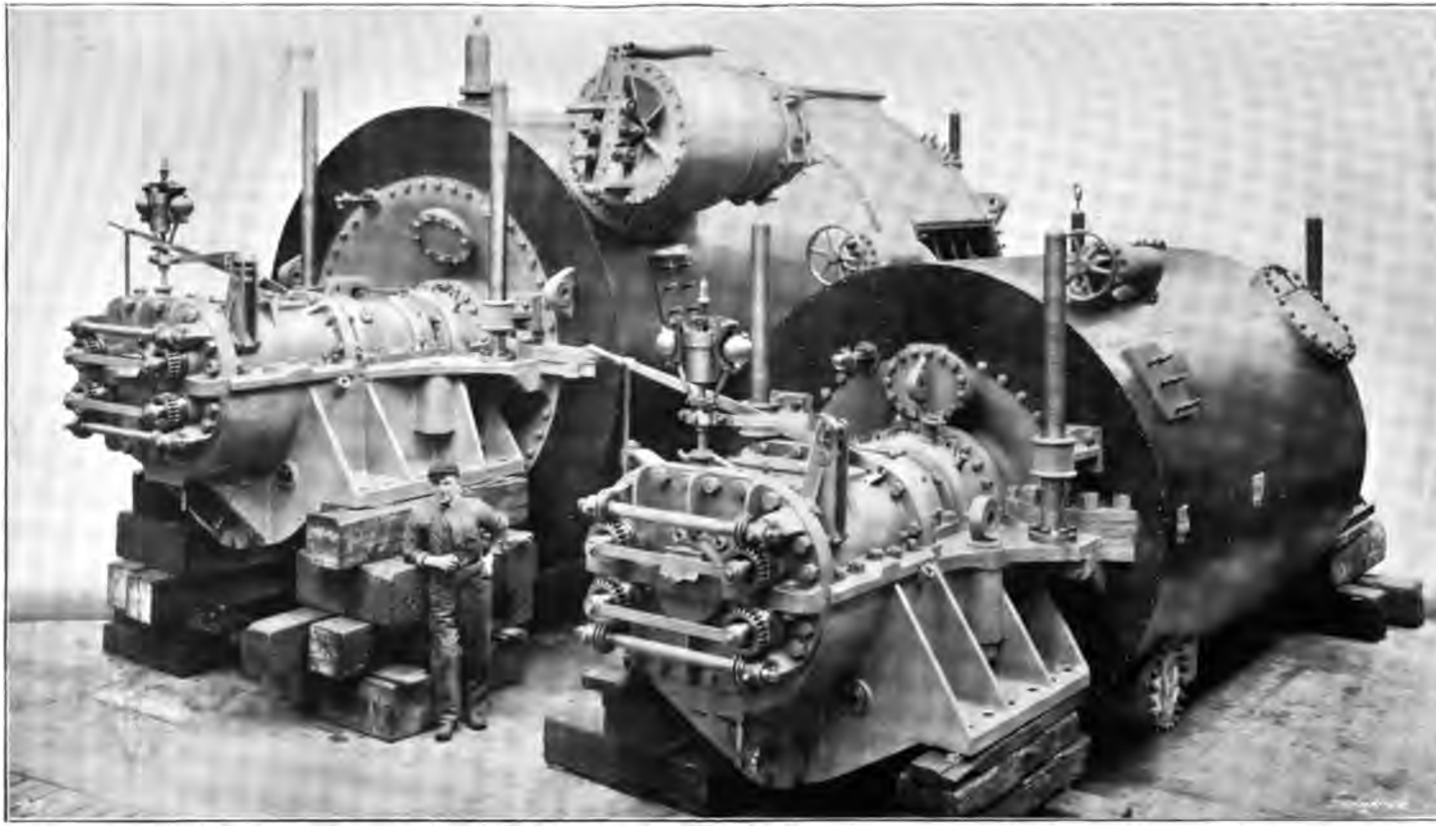
Oil Service to Bearings.

1. Oil Inlet. 3. Thermometer for Oil Outlet Temperature.
2. Sight Feed Glass. 4. Oil Drain.
5. Inspection Door.

the oil supply cease, rapid heating up and melting out of the main bearings would result. This, again, by the subsequent wear-down, might produce stripping of the vanes of the rotor and drum. As before mentioned, a sight glass, through which the oil passes, is often fitted, or a small cock with a hole bored out through the shell so that the oil spurts out when the cock is turned.

It is advisable, however, not to use the same oil for too long a period, as oil repeatedly used has its lubricating properties somewhat impaired, so that fresh supplies of unused oil become a necessity.

Difficulty is occasionally found in keeping down the temperature of the oil supply, which, if once heated up, has a decided tendency



H.P. Turbine and one L.P. Turbine of S.S. "Tenyo Maru" (Parsons' Turbines).

(Oriental Steamship Co. of Japan.)

Showing the Screw Adjusting Gear, Governor, Bye-Pass Valves on H.P., Non-Return Valve on L.P., Micrometer Dummy Gauges, and Relief Valves, &c.

H.P. Drum, 76 in. diameter; L.P. Drums, 106 in. diameter; Reverse Drums, 87 in. diameter.

Speed on Measured Course Trials, 20.62 knots.

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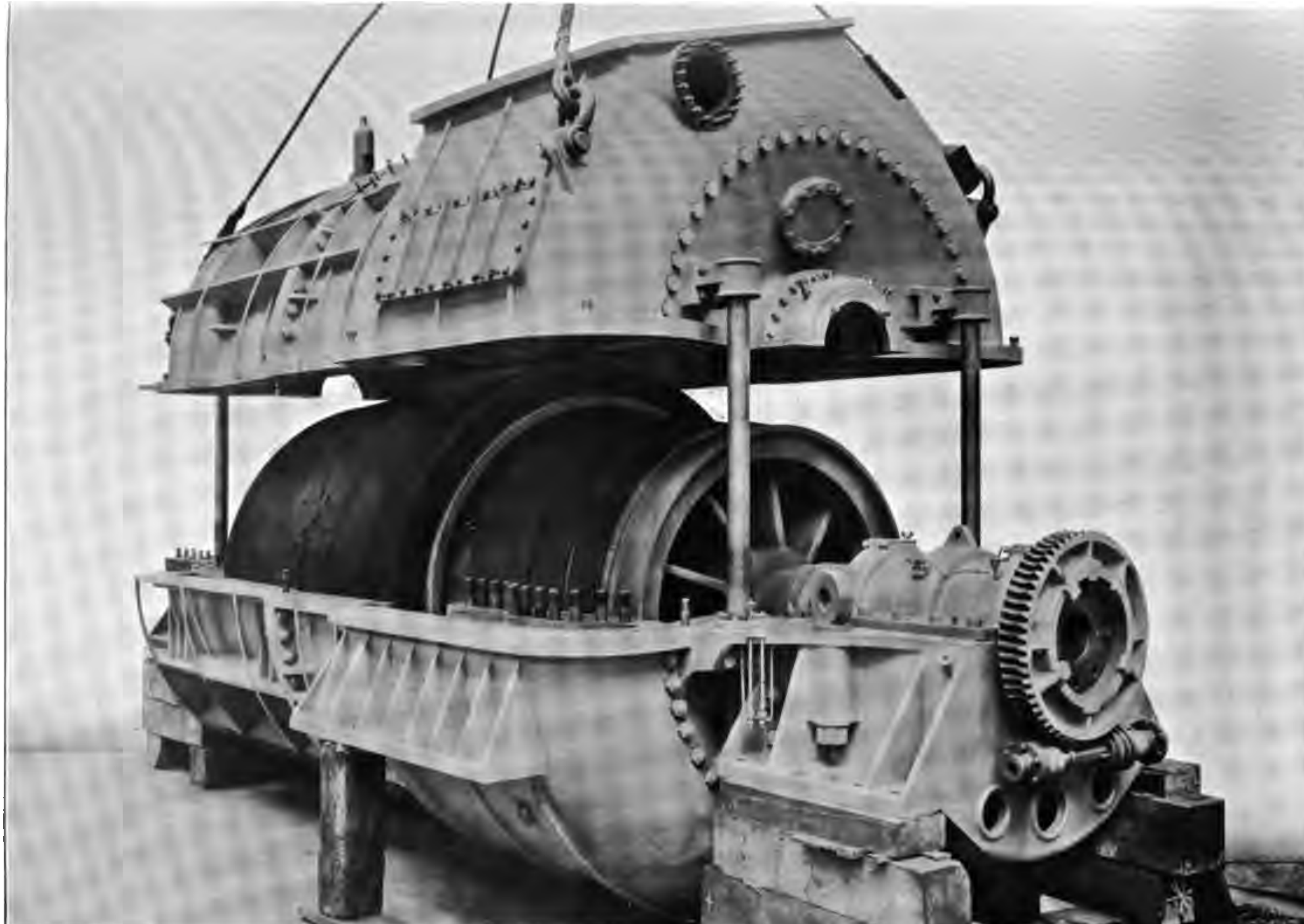

L.P. Turbine and Reverse Turbine, with Cover Raised (Parsons' Turbines). (S.S. "Tenyo Maru.")
Showing Reverse Wheel, Bolts of Reverse Casing, Turning Wheel and Gear, and Access Door on Upper Half Casing, &c.

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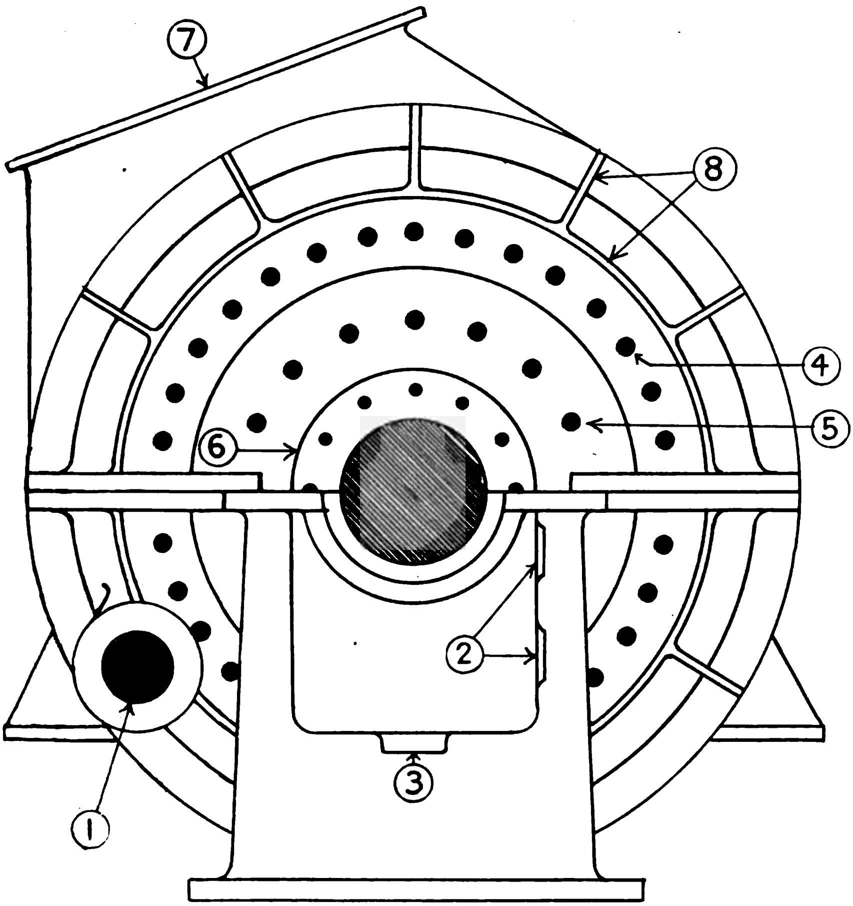
to remain very warm, the cooling water having little apparent effect in lowering the temperature.

NOTE.—The specific heat of oil is .36.

Oil is only used in the bearings at each end of the turbines,



End View (Aft) of L.P. Turbine Casing.

1. Astern Steam Inlet.
2. Oil Connections.
3. Oil Drain.
4. Bolts of Astern Casing.
5. Bolts of Astern Dummy Casing.
6. Gland Case.
7. Exhaust to Condenser.
8. Stiffening Webs.

and is prevented from entering the turbine casing by the oil baffle rings or "deflectors." The condensed steam, therefore, contains less oily matter than usual, with the resulting advantages to the boiler of less danger of collapsed furnaces and reduced pitting and corrosion.

SECTION III.

DATA FROM ACTUAL PRACTICE.

No. 1. DATA.

Steam flow = 4000 lbs. per minute.
Mean revolutions = 340.
H.P. rotor diameter = 63 in.
L.P. rotor diameter = 90 in.
Constant for blade rows = 1500000.
Constant for H.P. initial steam velocity = 3000.
Initial pressure = 172 lbs. gauge.

Then, V_t (H.P.) $= \frac{63 \times 3.1416 \times 340}{60 \times 12} = 93.4$, say 94 ft. per sec.

No. of blade rows $= \frac{1500000}{94^2} = 169$ (total).

H.P. turbine blade rows $= 169 \div 3 = 56$ rows.

Therefore $56 \div 4$ expansions = 14 blade rows for each H.P. expansion.

Again, V_t (L.P.) $= \frac{90 \times 3.1416 \times 340}{60 \times 12} = 133.5$ ft. per sec.

No. of blade rows $= \frac{1500000}{133.5^2} = 84$.

L.P. turbine blade rows $= \frac{84 \times 2}{3} = 56$.

Therefore $56 \div 8$ expansions = 7 blade rows for each L.P. expansion.

H.P. initial steam velocity $= \frac{3000}{\sqrt{169}} = 230$ ft. per sec.

Initial steam volume = 2.42 cub. ft. per lb. (187 lbs. absolute pressure).

Cubic feet flow per sec. $= \frac{4000 \times 2.42}{60} = 161.3$.

Required clear area through blades $= 161.3 \div 230 = .701$ sq. ft.

Annulus area $= .701 \times 3 = 2.103$ sq. ft.

" " $= 2.103 \times 144 = 302.832$ sq. in.

Then, diameter over blades $= \sqrt{\frac{63^2 \times .2854 + 302.832}{.7854}} = 65.9$, say 66 in.

And blade heights (1st expansion) $= \frac{66 - 63}{2} = 1\frac{1}{2}$ in.

Assuming a blade height ratio of 1.42 the blade heights of the other expansions may be found as follows:—

2nd expansion, $1.5 \times 1.42 = 2.13$, say $2\frac{1}{4}$ in. blade height.
3rd „ $2.25 \times 1.42 = 3.19$, „ $3\frac{1}{4}$ in. „ „
4th „ $3.25 \times 1.42 = 4.61$, „ $4\frac{1}{2}$ in. „ „

L.P. Blades.

Ratio of rotor diameters $= 90 \div 63 = 1.43$ (nearly).

$$\text{Blade heights (1st expansion)} = \frac{4.5 \times 1.42}{1.43 \times 1.43 \times 2} = 1\tfrac{1}{2} \text{ in.}$$

The blade lengths at the 2nd, 3rd, and 4th expansions will be the same as for the H.P., that is $2\frac{1}{4}$ in., $3\frac{1}{4}$ in., and $4\frac{1}{2}$ in.

5th expansion $= 4.5 \times 1.42 = 6.39$, say $6\frac{1}{2}$ in.
6th „ $= 6.39 \times 1.42 = 9.07$, „ 9 in.

The blades of the 7th and 8th expansions are also 9 in. long, but these blades are of the "wing" type which allow of a much larger exit area than the normal blades of the other expansions.

Steam Pressures and Volumes.

Initial Pressures, &c., at H.P. Expansions.

No.	Gauge Pressure.	Absolute Pressure.	Specific Volume in Cub. Ft.	Actual Volume in Cub. Ft.	Dryness.
1	172 lbs.	187 lbs.	2.420	2.400	1
2	112 „	127 „	3.462	3.402	.982
3	68 „	83 „	5.167	4.968	.961
4	43 „	58 „	7.238	6.860	.947
Terminal Pressure. 4	28 „	43 „	9.590	9.00	.938

Initial Pressures, &c., at L.P. Expansions.

No.	Gauge Pressure.	Absolute Pressure.	Specific Volume in Cub. Ft.	Actual Volume in Cub. Ft.	Dryness.
1	25 lbs.	40 lbs.	10.267	9.595	.934
2	12 „	27 „	14.863	13.698	.921
3	3 „	18 „	21.766	19.742	.907
4	6 in. vac.	12 „	31.879	28.400	.891
5	14 „	8 „	46.680	40.940	.877
6	18 „	6 „	61.201	52.987	.865
7	22 „	4 „	89.632	76.608	.854
8	25 „	2.5 „	139.488	117.711	.843
Terminal Pressure (Condenser). 8	27 „	1.5 „	225.580	186.045	.824

NOTE.—The "actual steam volumes" are calculated from the "error diagram" of Mr E. M. Speakman, and on referring to the diagram it will be noticed that the blade height curve approximates to that of the actual steam volumes at the last L.P. turbine expansions.

Ratio of Volumes.—At the various expansions the ratio of specific steam volumes is nearly the same as the ratio of blade heights, which the following will make clear.

H.P. Turbine.—At the 1st and 2nd expansions the specific steam volumes are 2.400 cub. ft. and 3.46 cub. ft.

Then, $3.46 \div 2.42 = 1.42$ ratio.

L.P. Turbines.—At the 6th and 7th expansions the specific steam volumes are 61.20 cub. ft. and 89.632 cub. ft.

Then, $89.632 \div 61.20 = 1.46$ ratio.

A similar ratio maintains between the respective actual volumes.

Number of Steam Expansions.—The total number of times the steam expands throughout the turbines *by pressures* is found by dividing the H.P. initial absolute pressure by the L.P. terminal absolute pressure.

Therefore, Number of expansions by pressures $= 187 \div 1.5 = 124$ times.

To find the number of steam expansions by volumes divide the L.P. terminal actual volume by the H.P. initial volume.

Therefore, Expansions by volumes $= 186.045 \div 2.4 = 77.5$ times.

Ratio of V_t to V_s.—The ratio of blade speed to steam speed at any expansion may be determined as follows:—

First H.P. Expansion.

Blade speed, $V_t = \dfrac{(63 + 1.5) \times 3.1416 \times 340}{60 \times 12} = 95.6$ ft. per sec.

Steam speed, $V_s = 230$ ft. per sec.

Ratio $= 95.6 \div 230 = .415$.

Fourth H.P. Expansion.

$$V_t = \frac{(63 + 4.5) \times 3.1416 \times 340}{60 \times 12} = 100 \text{ ft. per sec.}$$

Steam flow per sec. $= \dfrac{4000 \times 6.86}{60} = 457.3$ cub. ft.

Clear area between blades $= \dfrac{(63 + 4.5) \times 3.1416 \times 4.5}{3 \times 144} = 2.2$ sq. ft.

Then, $V_s = 457.3 \div 2.2 = 208$ ft. per sec.

Ratio, V_t, $V_s = 100 \div 208 = .48$.

First L.P. Expansion.

Blade speed, $V_t = \dfrac{(90 + 1.5) \times 3.1416 \times 340}{12 \times 60} = 135.7$, say 136 ft. per sec.

Steam flow per sec. $= \dfrac{2000 \times 9.595}{60} = 319.8$, say 320 cub. ft.

Clear area between blades $= \dfrac{91.5 \times 3.1416 \times 1.5}{3 \times 144} = .99$, say 1 sq. ft.

Then, $V_s = 320 \div 1 = 320$.
Ratio, V_t, $V_s = 136 \div 320 = .42$.

Fifth L.P. Expansion.

Blade speed, $V_t = \dfrac{(90 + 6.5) \times 3.1416 \times 340}{12 \times 60} = 143$.

Steam flow per sec. $= \dfrac{2000 \times 40.94}{60} = 1364.6$ cub. ft.

Clear area between blades $= \dfrac{(90 + 6.5) \times 3.1416 \times 6.5}{3 \times 144} = 4.56$ sq. ft.

Then, $V_s = 1364.6 \div 4.56 = 299$ ft. per sec.
Ratio, V_t, $V_s = 143 \div 299 = .48$ nearly.

NOTE.—In the foregoing calculations the actual volume of the steam has been taken at each expansion, and normal blades assumed, giving about one-third annulus area opening for steam flow.

Water Condensed in Turbines.—The water condensed by the nature of the expansion and drained off from the L.P. turbines per hour by the "wet" air pumps may be approximated as follows :—

Dryness at last L.P. expansion $= .824$.
Water condensed in turbines per min. $= 4000 \times (1 - .824) = 704$ lbs.
Water condensed per hour $= 704 \times 60 = 42240$.

Heat Drop and Horse-Power.

H.P. Initial 187 lbs. absolute $= 376°$ Temp. Fahr. Latent heat $= 847.6$.
$376 + 461 = 837$ absolute Temp. Dryness $= 1$.
L.P. Terminal 1.5 lbs. absolute $= 116°$ Temp. Fahr. Latent heat $= 1033.2$.
$116 + 461 = 577$ absolute Temp. Dryness $= .824$.
Then, Heat drop $= 847.6 \times 1 - 1033.2 \times .824 + 837 - 577 = 256.25$ B.T.U. per lb.
The mean blade efficiency assumed as 73 per cent. or .73.

Therefore, Horse-Power $= \dfrac{4000 \times 256.25 \times 778 \times .73}{33000} = 17640$ Horse-Power.

The equivalent I.H.P. $= 17653 \div .9 = 19600$.

NOTE.—The ratio of brake or shaft horse-power to I.H.P. is about .9 to 1.

Pressure Drops.—The following are the pressure drops which occur in each expansion of the foregoing case: assuming that the

terminal pressure of one expansion is the initial pressure of the next expansion, which is approximately correct:—

H.P. Expansions.	Absolute Initial Pressure.	Absolute Terminal Pressure.	Difference, or "Drop."
1	187 lbs.	127 lbs.	60 lbs.
2	127 "	83 "	44 "
3	83 "	58 "	25 "
4	58 "	40 "	18 "
...	...	...	147 lbs. total drop.

L.P. Expansions.	Absolute Initial Pressure.	Absolute Terminal Pressure.	Difference, or "Drop."
1	40 lbs.	27 lbs.	13 lbs.
2	27 "	18 "	9 "
3	18 "	12 "	6 "
4	12 "	8 "	4 "
5	8 "	6 "	2 "
6	6 "	4 "	2 "
7	4 "	2.5 "	1.5 "
8	2.5 "	1.5 "	1 "
...	...	...	38.5 lbs. total drop.

Mean pressure drop per blade row in H.P. turbine = 147 ÷ 56 rows = 2.62 lbs.
" " " " each L.P. " = 38.5 ÷ 56 " = .68 "

No. 2. Description and Trials of U.S.S. "Chester."

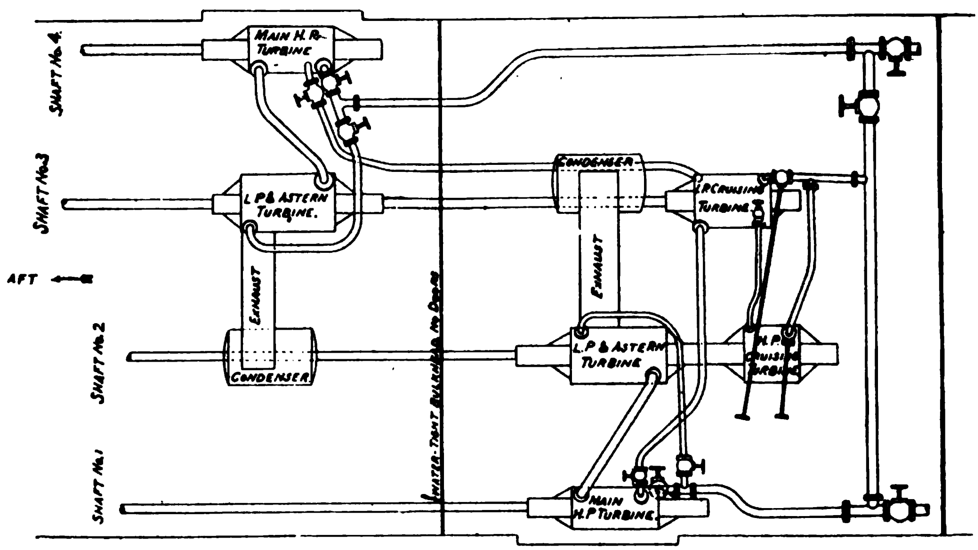
The writer is indebted to the President and Council of the American Society of Naval Engineers for permission to reproduce the following descriptions and trial data of the "Chester," from the pages of the *Journal* issued quarterly by the Society.

Displacement	3,750 tons.
I.H.P. (designed) at 24 knots	16,000.
Length (b. p.)	420 ft. 0 in.
" (over all)	423 ft. 2 in.
Beam	47 ft. $1\frac{1}{8}$ in.
Draught	16 ft. $8\frac{9}{16}$ in.
Ratio of length to beam	8.9
Number of screws	4
Wing shafts incline up and forward	2.46 deg.
Centre " " " "	1.44 deg.
Shafts diverge from centre line of ship	.72 degree.

NOTE.—The horse-power developed at the limit speed was much in excess of the 16,000 horse-power designed.

[illegible]

The propelling machinery consists of six Parsons' marine turbines driving four independent shafts, each shaft being fitted with one propeller; two cruising turbines are fitted to give fan economy at low speeds and powers. Exclusive of the cruising turbines there are two main high-pressure turbines exhausting into two low-pressure turbines, and in each of the latter there is incorporated a reversing turbine. To obtain economy at low and moderate speeds the six turbines may be used in three combinations.



Turbine Arrangement.
U.S. Cruiser Scout "Chester."

(1) **Low Speeds** (up to 18 knots).—The steam passes through all six ahead turbines, both the H.P. cruising and the I.P. cruising being connected up with the four main turbines. Steam admitted to the H.P. cruising turbine exhausts into the I.P. cruising turbine, and from the latter exhausts through separate pipes to each of the main H.P. turbines. From these latter it exhausts into the L.P. turbines and then into the main condensers.

(2) **Moderate Speeds** (up to 23 knots).—The steam passes through five ahead turbines, steam being admitted to the I.P. cruising turbine and passing thence to the two main high-pressure turbines, and from each of them to the connected low-pressure turbine. The high-pressure cruising turbine revolves idly in a vacuum.

(3) **Highest Speeds.**—Only the four main turbines are used, steam being admitted to each main H.P. turbine. Both cruising turbines revolve idly in a vacuum.

Reduction of power in each of the arrangements is obtained by throttling. Increased power may be obtained by admitting live steam to the I.P. cruising turbine in the *first* arrangement and to the main H.P. turbine in the *second* arrangement. Bye-pass valves are fitted from the 1st to the 2nd expansions in both main H.P., and auxiliary exhaust steam may be admitted to the 2nd expansions in main H.P. turbines and to both L.P. receiver pipes.

The turbines were designed for an initial working pressure of 250 lbs. per sq. in., to drive the shafts at 502 revolutions per minute for the contract speed of 24 knots.

Description of Propelling Machinery.

The propelling turbines are six in number, an H.P. cruising, I.P. cruising, two main high pressure and two low pressure, with reversing turbines incorporated into the exhaust ends of each of the latter. The reversing turbines revolve idly in the exhaust casing of the L.P. turbine when the engines are running ahead. The turbines are located in two compartments separated by an athwartship, watertight bulkhead. The inboard shafts turn outboard and the outboard shafts turn inboard.

The turbine cylinders are parted horizontally in the plane of the shaft, and the two halves strongly bolted together. The lower half is cast with extensions forward and aft, box shaped, for retaining the journal and thrust bearings. The cylinders are supported by feet, certain of which are securely bolted to the foundations and others have slotted holes for expansion. The feet are at a point directly under the journal bearings. The cylinders contain supports for the spindle and thrust-bearing brasses with oil pockets, also the lower halves of spindle glands, and have pockets under these glands for steam leak-offs. They have internal facings at their ahead steam ends for bolting on the dummies. Cylinder blading is caulked into grooves on the inside of the cylinder castings; these portions of the cylinders have been accurately bored and finished in a horizontal position. Hydrostatic tests were first applied to the cylinders, and subsequently a "baking" under moderate steam pressure for forty-eight hours to relieve internal stresses in the castings. The following table shows arrangement and dimensions of blading.

The rotors were statically balanced on two truly-levelled rails and the balance adjusted by removing metal, as necessary, from "chipping strips" left on the arms of the rotors for the purpose. The dynamic balancing was done by revolving the rotors under steam. At the same time dummy ring faces were worn down, and bearings adjusted.

Blading.

Expansions.	Rows.	Heights, Inches.	Pitch, Inches.	Clearances, Inches. (Cold.)
H.P. Cruising Turbine.				
1st	12	$\frac{3}{8}$	$\frac{7}{8}$	.03
2nd	12	$\frac{1}{2}$	$\frac{7}{8}$	.035
3rd	12	$\frac{5}{8}$	1	.04
I.P. Cruising Turbine.				
1st	15	$1\frac{1}{8}$	$1\frac{1}{4}$	.04
2nd	15	$1\frac{7}{16}$	$1\frac{1}{4}$	.04
3rd	15	$1\frac{3}{4}$	$1\frac{5}{16}$	.045
High-Pressure Turbines.				
1st	12	$\frac{7}{8}$	$1\frac{1}{4}$	.03
2nd	12	$1\frac{1}{4}$	$1\frac{1}{4}$	.035
3rd	12	$1\frac{3}{4}$	$1\frac{5}{16}$	.04
4th	12	$2\frac{1}{2}$	$1\frac{3}{8}$	.045
5th	12	$3\frac{1}{2}$	$1\frac{7}{16}$	.05
6th	10	5	$1\frac{13}{16}$	.05
Low-Pressure Turbines.				
1st	5	$2\frac{1}{2}$	$1\frac{5}{16}$	.055
2nd	5	$3\frac{1}{2}$	$1\frac{3}{8}$	.06
3rd	4	5	$1\frac{13}{16}$	.07
4th	3	7	$1\frac{3}{4}$	.08
5th	3	7	$1\frac{15}{16}$	.08
6th	4	7	$2\frac{5}{16}$	.08
7th	4	7	$2\frac{5}{16}$	.08
Astern Turbines.				
1st	6	$\frac{1}{2}$	$1\frac{1}{8}$	.045
2nd	6	1	$1\frac{3}{8}$	.05
3rd	6	2	$1\frac{3}{8}$	.06
4th	6	2	$1\frac{1}{2}$	.06
5th	6	2	$1\frac{1}{2}$	.06

Turbine Cylinder Data.

Cylinder.	Diam. of Rotor Drum.	Length of Rotor Drum.	Diameter of Cylinder for each Expansion.						
			1st.	2nd.	3rd.	4th.	5th.	6th.	7th.
	In.	In.	In.	In.	In.	In.	In.	In.	In.
H.P. cruising - - - -	60	36	60¾	61	61¼	...	...	...	...
I.P. cruising - - - -	49	60.5	51¼	51⅞	52.5	...	...	...	...
Main H.P. port - - - -	42	103.5	43¾	44.5	45.5	47	49	52	...
„ „ starboard - -	42	103.5	43¾	44.5	45.5	47	49	52	...
„ L.P. port - - - -	65	53⅜	70	72	75	79	79	79	79
„ „ starboard - -	65	53⅜	70	72	75	79	79	79	79
Astern, port - - - - -	50	43	51	52	54	54	54	...	...
„ starboard - - -	50	43	51	52	54	54	54	...	...

NOTE.—The diameter of cylinder at each expansion is obtained by adding twice the blade height to the rotor drum diameter.

"The blades were installed in accordance with the Parsons system. They were cut to length, saw-cuts made where required for binding, caulking grooves stamped into the base of the blades, all done by the same machine, and the blades then caulked into place. Caulking strips having curves corresponding to the blades were cut to length and placed alternately at the base of each blade. The blades and caulking strips were alternately tightened into the groove by means of a caulking tool, after which the binding wire was inserted and the lacing put on around binding wire and blades with silver solder. Blades were turned up in the vertical line before the insertion of the binding wire, and were finally filed up to remove burrs.

"The blades are of a composition consisting of 72 per cent. copper and 28 per cent. zinc, the caulking strips 63½ per cent. copper and 36½ per cent. zinc, the binding strip 72 per cent. copper and 28 per cent. zinc, and the lacing wire pure copper. Blades having binding strips are soldered to the latter with a silver solder. Steam supply and exhaust piping is as shown on page 143. In the exhaust pipe from H.P. cruising turbine to I.P. cruising turbine, and in the two exhaust pipes from the I.P. cruising to the main H.P. turbines, spring-loaded, self-closing valves are installed to isolate the cruising turbines while running at higher rates of speed and while manœuvring. A 3½-in. spring-relief valve for the H.P. cruising turbine is installed in its exhaust pipe, set at 125 lbs., and one for the I.P. cruising turbine is installed on after end of its upper casing, set at 160 lbs. A governor, arranged to pull out a pawl and close a butterfly valve in the main steam pipe, is fitted on each shaft, the object being merely to shut down automatically in the event of accident. An unusual feature in piping is the low-pressure exhaust pipe, which is an elbow pipe rectangular in cross section, 3 ft. 9 in. by 4 ft. 6 in. and of ⅜-in. steel plate. The dummies in the steam ends of turbines for preventing steam

from leaking from steam belts direct to the inside of drums consist of a series of rectangular grooves in the spindle wheels against the side of each of which a brass strip lies closely without touching. These brass strips are caulked into an annular iron casting made in halves and bolted to the inside of the cylinder. A micrometer for measuring the opening of dummies between brass strips and running steel faces is fitted on each cylinder. Steam glands are fitted around turbine spindles at cylinder ends, packed with labyrinth stuffing boxes. These glands are to obviate, in the case of high-pressure turbines, the leakage of steam from the casings, and in the case of the low-pressure turbines, the leakage of air from the atmosphere into the casing and thence to the condenser. There are rows of brass strips let into the shaft and the cast-iron gland sleeve, with thin edges just clearing the opposite member. Snap rings of composition H are fitted at the outer end of each gland box, and just inside of these rings connection is made with an equaliser pipe joining all similar pockets, and a valve is provided from auxiliary exhaust line for maintaining the pressure in this pipe. Strainers of the basket type in accordance with current turbine practice, made with bodies and covers of composition G, and with strainer baskets of sheet brass, are installed, one in the live-steam supply to each turbine and one in the auxiliary exhaust connection to the 2nd expansion of the main H.P. turbines and L.P. turbines. A system of guide rods is installed for lifting turbine casings, one on each of the four corners, placed vertically and graduated in inches, to permit of the even lifting of the covers. Special portable guides are supplied for guiding the spindles when lifting them, to prevent the stripping of blades. Turning gear is fitted at the after end of each turbine for turning it by hand. It consists of a worm wheel on the shaft, meshing with a worm operated by a ratchet wrench. Suitable lifting gear is provided for lifting casings, spindles, &c., consisting of overhead trolleys on frames, with chain falls and slings."

Shafting and Shaft Bearings.—"The shafting is of Class 'A' forgings and is solid. The thrust shaft is on spindle ends. The stern-tube shaft and propeller shaft is in one piece, inboard sections 48 ft. 3 in. long, and outboard lengths 46 ft. $9\frac{11}{16}$ in. long, in both cases $8\frac{1}{2}$ in. diameter. The line shafting is 8 in. in diameter. There are two shaft bearings to each turbine—twelve in all—those for the cruising turbines being 10 in. long, and those for the main turbines 15 in. The bearing shells, of brass, are semicircular, and are lined with white metal. Broad strips of the bearing shell are exposed slightly below the line of the white metal to catch the spindle in case the white metal gives out. The bottom half can be rolled out without lifting the spindle, the top half will lift off. These bearings are lubricated by oil supplied under pressure. There are six thrust-shaft bearings, those for the cruising turbines having eight collars and rings, those for main turbines fifteen collars and rings. These bearings are so constructed that the longitudinal

position of the shaft and the dummy clearance can be readily adjusted. The thrust bearings are in two halves. The steam balance is so arranged as to be practically equal to the thrust of the propeller at all speeds. The thrusts are lubricated by oil under pressure. Oil supplied to thrust bearings and journal bearings is collected in a chamber in the box ends of lower cylinder castings, and drains by gravity into a return pipe. There are fifteen line-shaft bearings each 12 in. long, lined with white metal, and with ends provided with oil baffles; provision is made for their lubrication by oil under pressure. Water service to a jacket around oil chambers is provided only in case of line-shaft bearings, the water service pipe discharges into stern-tube stuffing box, and has a branch pipe for draining the latter. Water service by hose can be provided in emergency. Stern-tube bearings for each shaft consist of a bearing in each end of the tube in each case. They are each 51¾ in. in length. Each shaft also has one strut bearing 51¾ in. in length, and fair-water sleeves are fitted on the forward side of the strut at one end, and on the after side of the stern tube at the other. There are sixteen bulkhead stuffing boxes where shafts pass through watertight bulkheads.

"An expansion coupling is fitted between each L.P. turbine and the cruising turbine on same shafting. The forward end of each L.P. turbine shaft is built with twelve teeth, which engage with a slotted collar bolted to the flange coupling on after end of cruising turbine shafting. This permits of disconnecting cruising turbines, as well as of expansion. As but slight increase efficiency is gained by disconnecting, this is not done as a rule, the cruising turbines being run idle in a vacuum when not in use."

Propellers.—There are four solid, true-screw, manganese bronze propellers, each having three blades; diameter, 6 ft.; pitch, 6 ft.; area, projected, 17.02 sq. ft.; helicoidal, 19 sq. ft.; disc area, 28.27 sq. ft.; immersion of upper tip of blade, 69½ in. inboard, 57½ in. outboard. The driving surfaces of all propeller blades were made a true surface by machining. The wing propellers are ahead of the inner propellers, and the former turn inboard and the latter outboard, though it was originally intended that all four turn outward.

Main Condensers.

There are two main condensers, one in each engine-room, located inboard and abreast of each L.P. turbine. They are horizontal and cylindrical, of the surface-condensing type. The shells are made of Class "B" boiler plate, water chests of composition, tubes of composition, and tube sheets and supporting plates of "rolled naval bronze." The forward water chest, being the one for the entrance and exit of circulating water, has a division plate with a 7-in. bye-pass valve on outside of chest. Each water chest has eight 10-in. man-

holes fitted with composition covers, inside of which are placed zinc plates. The following is the data for each :—

Cooling surface	8,999 sq. ft.
Number of tubes	5,630
Tube thickness	18 B.W.G.
Outer diameter of tubes	$\frac{5}{8}$ in.
Length of tubes	10 ft. $0\frac{1}{4}$ in.

Augmenter Condenser.

An arrangement called a "vacuum augmenter," devised by the Parsons Turbine Company, is installed for the purpose of increasing the vacuum above that obtained by the air pump.

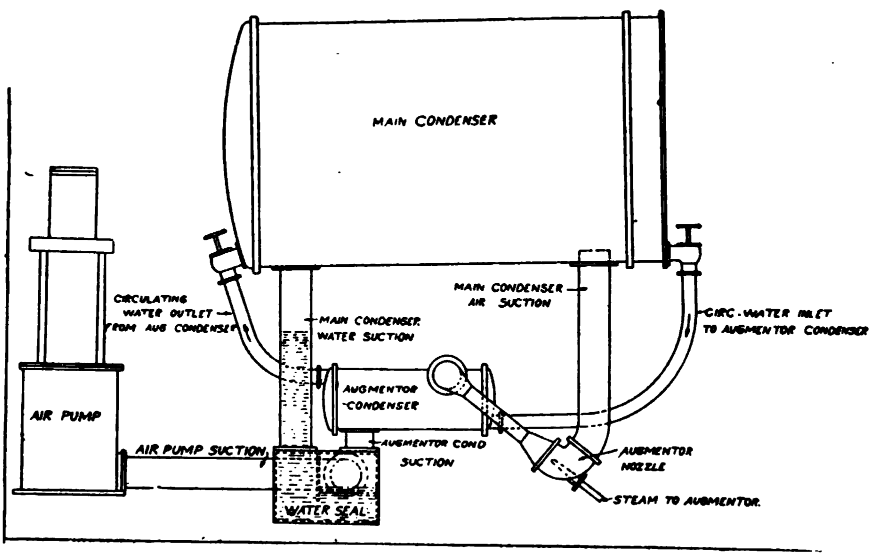


FIGURE 3

Parsons' Vacuum Augmenter.

The augmenter consists of a steam syphon drawing air from the condenser and discharging it to the air pump suction at a pressure from 1 to $1\frac{3}{10}$ in. of mercury higher than the condenser pressure. The discharge from the syphon passes through a small condenser in order to condense the steam of the syphon jet. The air pump has a direct water suction from the condenser through a pipe having a water seal and holding a head of water equal to the difference in pressure produced by the augmenter jet. The augmenter was in use at all times during the trials, and produced an increase of vacuum of from 1 to $1\frac{1}{4}$ in.

Lubrication.

The lubrication of all main journal bearings, thrust bearings, and line-shaft bearings is by oil supplied under about 10 lbs. pressure by steam-driven oil pumps. The oil passes through a cooler on its way to the bearings. Tanks of 150 gallons capacity are located in the lower parts of engine-rooms, one in each, from which the oil pumps draw. These tanks take the return of the oil from all bearings supplied with forced lubrication, by gravity. The discharge of the oil from turbine bearings is through glasses so that the flow can be observed, and thermometers are fitted to these discharges. The oil pipes are all of copper, and have no valves fitted either to or from the bearings. Two pumps, Blake, vertical, simplex-piston type, $10 \times 9 \times 12$, are supplied, one in each engine-room. Either or both may be used. An oil cooler of the surface-condenser type, with oil passing through the tubes and the water around them, is provided in each engine-room. The oil passes four times the length of the cooler by means of bridges in the oil heads. The circulating water is taken from either the engine-room fire and bilge pumps or main circulating pumps, and discharges through outboard delivery pipes. Additional oil may be supplied to the forced lubrication system by an overhead gravitation system from the tanks containing reserve oil. The outlets for oil in the casings are at a high level, so that there is always in the casing wells a large gathering of oil which is at about the same temperature as the shaft.

Main Air Pumps.

Two independent Blake, vertical, twin-beam air pumps, $14 \times 35 \times 21$, are installed, one for each engine-room. The suction openings are 12 in. in diameter, and the discharge 11 in. The air-pump suctions take from the lower end of the vacuum-augmenter condensers and water seals (see Vacuum-Augmenter Condenser and Fig. 3).

Main Circulating Pumps.

For each main condenser there is one main circulating pump, centrifugal, with vertical compound engine of B.I.W. design, steam cylinders, 10 in. and 16 in. in diameter; stroke, 10 in.; 42-in. diameter runner. Each pump is of sufficient power to discharge 11,000 gallons of water per minute at about 360 revolutions. Each pump is fitted with pipes and valves to draw from the sea and the main drain, and to deliver into the condenser or overboard through valves in the condenser water chest. Circulating water for oil coolers may be supplied from these pumps.

Boilers.

There are twelve Normand water-tube boilers of the "intermediate" type in the vessel in three watertight compartments, four to each compartment. The following is the boiler data :—

Designed working pressure	250 lbs.
Test pressure	400 lbs.
Ratio of grade surface to heating surface	1 to 46
Length of grates	7 ft. $2\frac{1}{8}$ in.
Width of grates	8 ft. $\frac{7}{8}$ in.
Per cent. of air space in grates	49
External height	12 ft. $5\frac{5}{8}$ in.
„ length	11 ft. $6\frac{5}{8}$ in.
„ width	11 ft. 4 in.
Number of furnaces	1
Grate surface	58 sq. ft.
Heating surface	2,670 sq. ft.
Total grate surface	696 sq. ft.
„ heating surface	32,040 sq. ft.
Weight of boiler, with water	19,675 tons
Boiler tubes	cold-drawn seamless steel.
Number of tubes each boiler	986, No. 10 B.W.G., O.D., $1\frac{3}{8}$ in. lengths vary for each row.
Mean height of smoke pipes above grates	63.25 ft.
Diameter of smoke pipes	72 in.

Performance.

Four Hours' Official Trial.

Steam Pressure.

	Starboard.	Port.
Mean steam pressure at engines, pounds	242.4	246.0
In main H.P. turbine (per gauge)	237.0	240.0
In 2nd I.P. cruising turbine (absolute)	237.0	240.0
In L.P. turbine (absolute)	30.2	29.9
Vacuum in condensers, inches of mercury, mean	28.9	28.3
Average revolutions per minute	614	
Speed of ship, in knots	26.522	
Slip of propeller, in per cent. of its own speed, based on mean pitch. Shafts are numbered from starboard to port.	(1) 25.71 (2) 25.99 (3) 26.18 (4) 24.34	
Air pressure in fire-rooms, in inches of water, mean	2.74	

Coal.

Kind and quality used on trial	Pocahontas, hand picked.
Pounds, per hour, main and auxiliary engines, during trial	38,332
Pounds of coal per square foot of grate surface, per hour	55.08

Twenty-four Hours' Official Trial at 22.5 Knots.

Steam Pressure.

	Starboard.	Port.
Mean steam pressure at engines, pounds	230.0	
In I.P. cruising turbine (gauge)	...	192.0
In main H.P. turbine (gauge)	96.0	93.0
In L.P. turbine (absolute)	15.0	13.8
Vacuum, in inches	28.3	29.5
Average revolutions per minute	473	
Speed of ship, in knots	22.782	
Slip of propeller, in per cent. of its own speed, based on mean pitch. Shafts are numbered from starboard to port.	(1) 15.66	
	(2) 16.94	
	(3) 26.08	
	(4) 15.15	
Air pressure in fire-rooms, in inches of water, mean	0.7	

Coal.

Kind and quality used on trial	Pocahontas, run of mine.
Pounds per hour, main and auxiliary engines, during trial	18,063
Knots per ton of coal	2.82
Pounds of coal per square foot of grate surface, per hour	25.95

Twenty-four Hours' Official Trial at 12 Knots.

Steam Pressures.

	Starboard.	Port.
Mean steam pressure in engine-room	148.0	
In H.P. cruising turbine (gauge)	77.3	
In I.P. cruising turbine (gauge)	...	34.6
In main H.P. turbine (gauge)	8.0	6.8
In L.P. turbine (absolute)	3.3	3.4
Vacuum, in inches	28.4	28.6
Average revolutions per minute	250	
Speed of ship, in knots	12.2	
Slip of propeller, in per cent. of its own speed, based on mean pitch. Shafts are numbered from starboard to port.	(1) 12.21	
	(2) 24.66	
	(3) 19.59	
	(4) 12.08	

Coal.

Kind and quality used on trial	Pocahontas, run of mine.
Pounds per hour, main and auxiliary engines, during trial	4,091
Knots per ton of coal	6.68

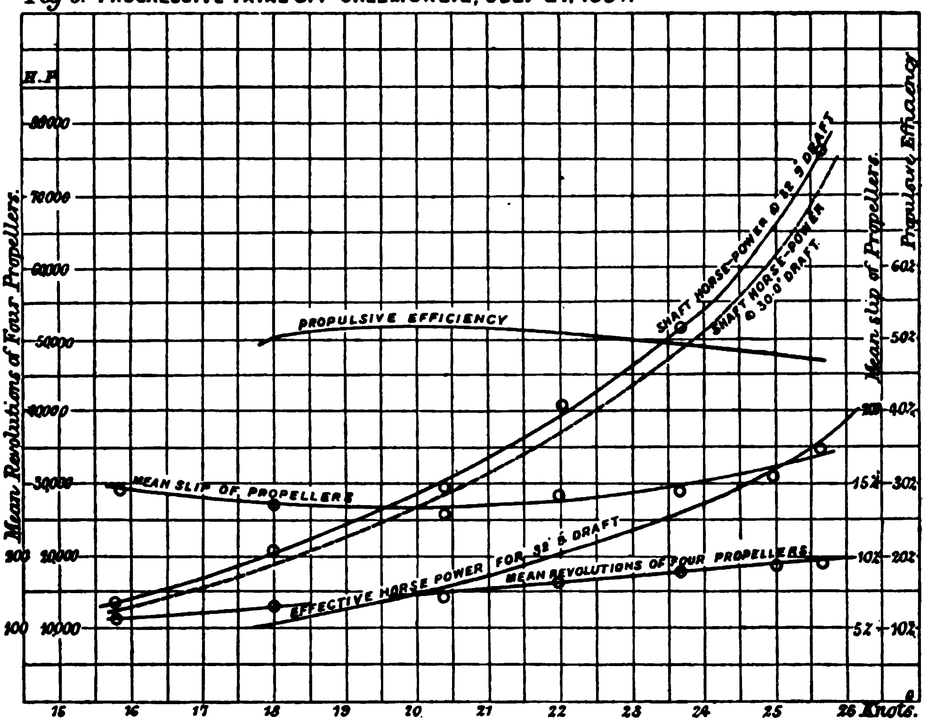
Deduced Data.

Pounds of coal per square foot of grate surface per hour	17.63

Note.—Vibration of turbines and hull were practically *nil* at all speeds, a slight tremor only being noticed in the after cabin at top speed, a propeller influence. Both bow waves and stern waves were remarkably small.

3. The following extracts from the paper read by Thomas Bell, Esq., before the Institution of Naval Architects, 9th April 1908, entitled "Speed Trials and Service Performance of the Cunard Turbine Steamer 'Lusitania,'" may prove of interest :—

Fig 6. PROGRESSIVE TRIAL OFF SKELMORLIE, JULY 27, 1907.



Speed Power Curve, "Lusitania."

ENGINES.

Turbines.	Diameter of Rotor.	Length of Blades. In First Expansion.	Length of Blades. In Last Expansion.
	In.	In.	In.
High-pressure	96	$2\frac{3}{4}$	$12\frac{3}{8}$
Low-pressure	140	$8\frac{1}{4}$	22
Astern	104	$2\frac{1}{4}$	8

Total cooling surface, main condensers	=	82,800 sq. ft.
Area of exhaust inlet	=	158 sq. ft.
Bore of circulating discharge pipes	=	32 in.
Diameter of tunnel shafts	=	20 in. external, 10 in. hole.
Diameter of propeller shafts	=	22 in. external, 10 in. hole.

Boilers.

Working pressure	195 lbs. per sq. in.
Twenty-three double-ended boilers	17 ft. 6 in. mean diam. by 22 ft. long.
Two single-ended boilers	17 ft. 6 in. mean diam. by 11 ft. 4 in. long.
Total number of furnaces	192.
Total grate surface	4,048 sq. ft.
Total heating surface	158,352 sq. ft.
Total length of boiler-rooms	336 ft.
Total length of main and auxiliary engine-rooms	149 ft. 8 in.

Table IV.

Time.	Pressures. H.P. Recr.	Pressures. L.P. Recr. Pounds.	Vacuum at 30 In. Barometer.	Revs. per Minute.	Speed in Knots.	Shaft Horse-power.	Slip of Propellers.
	Lbs.	Lbs.	In.				
First run	151	$2\frac{3}{4}$	28	191.3	25.62	72,500	...
Second run	152	$2\frac{3}{4}$	28	191.1	26.36	72,000	...
Third run	153	3	28	191.9	25.31	72,000	...
Fourth run	147	$2\frac{1}{2}$	28	191.0	26.16	72,100	...
Fifth run	149	$2\frac{1}{2}$	28	190.2	25.26	70,800	...
Sixth run	149	$2\frac{1}{2}$	28	191.2	25.95	71,800	...
Mean of means	150	$2\frac{3}{4}$	28	191.2	25.77	71,910	15.3 %

TABLE V.

Actual steam and coal consumption of main and auxiliary engines at various speeds under conditions prevailing on official trials, viz., turbo-generators exhausting to auxiliary condensers, other auxiliaries exhausting to heaters.

Shaft horse-power	13,400	20,500	33,000	48,000	68,850
Speed in knots	15.77	18.0	21.0	23.0	25.4
Temperature of feed water	200°	200°	199°	179°	165°
Total consumption of auxiliaries in lbs. per hour	71,000	85,700	76,400	96,700	116,500
Total consumption of turbines in lbs. per hour	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries in lbs. per turbine horse-power hour	5.3	3.72	2.6	2.01	1.69
Steam consumption of turbines in lbs. per horse-power hour	21.23	17.24	14.91	13.92	12.77
Total steam consumption in lbs. per horse-power hour	26.53	20.96	17.51	15.93	14.46
Coal consumption in lbs. per horse-power hour	2.52	2.01	1.68	1.56	1.43
Estimated coal consumption in tons on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, &c.	2,980	3,190	3,670	4,520	5,390

TABLE VI.

Estimated steam and coal consumption at various speeds, allowing for the additional auxiliary steam consumption found requisite under actual service conditions for the washing water supply, &c., with a full complement of passengers, weather conditions being as on official trial.

Shaft horse-power	13,400	20,500	33,000	48,000	68,850
Speed in knots	15.77	18.0	21.0	23.0	25.4
Temperature of feed water	200°	200°	200°	200°	200°
Total consumption of auxiliaries in lbs. per hour	93,500	100,900	112,700	127,500	149,700
Total consumption of turbines in lbs. per hour	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries in lbs. per turbine horse-power hour	6.97	4.92	3.41	2.65	2.17
Steam consumption of turbines in lbs. per horse-power hour	21.23	17.24	14.91	13.92	12.77
Total steam consumption in lbs. per horse-power hour	28.2	22.16	18.32	16.57	14.94
Coal consumption in lbs. per horse-power hour	2.76	2.17	1.8	1.62	1.46
Estimated coal consumption in tons on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, &c.	3,270	3,440	3,930	4,700	5,490

TABLE VII.

Abstract of Engine-Room Log for third voyage west: Queenstown to New York.

Date when last dry docked, 22nd July 1907. Mean draft, leaving Queenstown, 33 ft. 7 in. Mean draft, arriving New York, 30 ft. 10 in.

Date, 1907.	Steam Pressures.			Temperatures.		Vacuum.	Barometer.	Length of Day.		Distance by Observation.	Mean Speed.	Mean Revs.	Mean Slip.	Coal Consumed for Main and Auxiliary Engines.
	Boilers.	H.P. Recrs.	L.P. Recrs.	Hot-well.	Feed Water.									
	Lbs.	Lbs.	Lbs.			In.	In.	Hrs.	Mins.	Naut. Mls.			Per Cent.	Tons.
Noon, 3rd Nov.	170	140.0	2.3	68°	200°	28	30.4	0	52	21	24.24	182.5	16.5	40
Noon, 4th Nov.	160.1	142.2	2.2	78°	197°	28	29.7	24	57	606	24.28	182.6	16.4	1,090
Noon, 5th Nov.	167.3	140.6	2.3	78°	198°	28.2	30	25	2	616	24.6	182.8	15.4	1,090
Noon, 6th Nov.	168.3	140.4	2.5	70°	196°	28.2	30.1	24	55	618	24.8	183.5	15.1	1,090
Noon, 7th Nov.	168.3	138.3	2.2	72°	195°	28	29.6	24	52	610	24.52	181.4	15.0	1,090
1.14 A.M., 8th Nov.	165	132.5	1.5	75°	200°	27.8	29.3	14	2	310	22.09	174	20.2	*576
Means	168	139.3	2.2	74.5°	197°	28.1	29.8	Total. 114	40	Total. 2,781	24.25	181.1	15.9	4,976

* This includes all coal used till 10 A.M. on the 8th.

Summary of total coal consumed on voyage:—Liverpool to Queenstown, 408 tons; Queenstown to New York, 4,976 tons; galleys, &c., 18 tons—total coal taken from bunkers, from leaving landing stage, Liverpool, till moored at wharf, New York, 5,402 tons. Passage—Queenstown to Sandy Hook—4 days, 18 hours, 40 minutes.

TABLE VIII.

Date, 1907.	Length of Steaming Day.		Distance Run, Nautical Miles.	Speed, Knots.	Total Distance Steamed.	Total Time.		Mean Average Speed.
	Hrs.	Min.				Hrs.	Min.	
Noon, 3rd Nov.	0	52	21	24.24	21	0	52	24.24
Noon, 4th Nov.	24	57	606	24.28	627	25	49	24.27
Noon, 5th Nov.	25	2	616	24.6	1,243	50	51	24.44
Noon, 6th Nov.	24	55	618	24.8	1,861	75	46	24.57
Noon till midnight, 6th Nov.	12	30	315	25.2	2,176	88	16	24.65
Noon, 7th Nov.	12	22	295	23.85	2,471	100	38	24.55
Morning, 8th Nov.	14	2	310	22.09	2,781	114	40	24.25

"Table VI. has been compiled for comparison with Table V., to show the additional consumption of steam for auxiliary purposes under actual working conditions at sea with the ship full of passengers. This shows very clearly the demand which modern improvements make on the steam, and hence coal consumption of a large passenger vessel. An additional line has been added to Tables V. and VI. to show total coal consumption on a voyage of 3,100 nautical miles at the various speeds and under the different conditions.

"With reference to the third voyage west, from 2nd November to 8th November of last year, thanks to the courteous permission of the chairman of the Cunard Company, the leading particulars of the official engine-room log are summarised in Table VII. Regarding the mean draft of the vessel at sea, it may be remarked that, after the second day out, certain of the forward tanks were gradually filled for the purpose of avoiding excessive trim, so that the mean draft on 5th, 6th, and 7th November was approximately 32 ft., or very little more than the mean of the first pair of runs from Corsewall Point to the Longships and back. The conditions, however, were otherwise very different, for, with the exception of the twelve hours of fine weather and smooth sea from noon till shortly after midnight on 6th November, it was throughout the average mid-Atlantic winter weather—namely, strong winds and resulting boisterous sea. Up till midnight on the 6th, *i.e.*, for 2,176 out of a total of 2,781 nautical miles, the mean speed works out at 24.65 knots; but, unfortunately, early on the 7th the wind freshened, gradually increasing to a furious south-west gale, which reached its height about 4 P.M., and reduced the average speed for the last twenty-four hours below 23 knots, and thus brought down the mean average for the completed voyage to 24.25 knots. Table VIII., giving the mean average speeds at the different stages of the voyage, shows very clearly the effect of this gale, unfortunate so far as

preventing the vessel from complying with the contract conditions, but giving those connected with the ship an opportunity of thoroughly satisfying themselves as to her behaviour when driving through the huge waves at about 22½ knots, without any racing of engine or sign of labouring, and dispelling the idea, current in some minds, that turbine-propelled ships do not show to advantage in heavy weather.

" The following particulars of the steam consumption are given in conjunction with the figures of coal consumption set forth in Table VII. Throughout the voyage a careful record of the feed pump counters gave an average of 998,000 lbs. of water pumped into the boilers per hour. Of this, about 114,000 lbs. was used by auxiliary machinery exhausting into the feed heaters, 26,000 lbs. by the evaporating plant supplying feed make-up and washing water, and about 6,500 lbs. for steam to the thermotanks, galleys and pantries, both of which latter figures are based on data obtained from tests carried out before the vessel left the Clyde. Hence, taking the average shaft horse-power as 65,000, the steam consumption per shaft horse-power per hour works out as follows :—

	Total Water.	Per Shaft Horse-power Hour.
Main turbines - - - - -	851,500 lbs. =	13.1 lbs.
Auxiliary machinery - - -	114,000 lbs. =	1.75 lbs.
Evaporating plant and heating -	32,500 lbs. =	.5 lb.
	998,000 lbs.	15.35 lbs.

Average amount of coal burnt per hour for all purposes = 43½ tons.
Water evaporated per lb. of coal = 10.2 from a feed temperature of 196°.
Water evaporated per lb. of coal = 10.9 from and at 212°.
Coal for all purposes per shaft horse-power per hour = 1.5 lbs.
Coal per square foot of grate per hour = 24.1 lbs.

" Taking a mean displacement of 36,000 tons, this represents at 24½ knots a consumption of almost exactly 11 lbs. of coal per 100 nautical miles per ton of displacement. The coal used was half South Wales and half Yorkshire, practically the same as on the official trials."

4. Blade Tip Clearances taken Cold and Hot.

H.P. Turbine.

COLD.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
1	.042	.057	.042	.044
2	.042	.052	.044	.048
3	.050	.056	.050	.049
4	.059	.058	.071	.058

HOT.

1	.024	.048	.026	.036
2	.026	.046	.023	.036
3	.032	.048	.032	.037
4	.043	.053	.054	.050

P.L.P. Turbine.

COLD.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
1	.064	.078	.078	.065
2	.067	.081	.079	.066
3	.073	.092	.076	.068
4	.077	.093	.090	.083
5	.082	.106	.103	.098
6	.098	.108	.108	.105
7	.100	.108	.108	.106
8	.103	.108	.108	.108

HOT.

1	.055	.055	.069	.040
2	.053	.054	.070	.038
3	.060	.056	.058	.035
4	.062	.054	.068	.050
5	.072	.058	.073	.065
6	.076	.063	.084	.066
7	.068	.053	.075	.063
8	.068	.053	.060	.061

S.L.P. Turbine.

Cold.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
1	.059	.073	.c60	.067
2	.066	.077	.061	.073
3	.078	.088	.074	.084
4	.082	.089	.075	.086
5	.092	.096	.086	.088
6	.103	.103	.096	.095
7	.103	.103	.097	.093
8	.103	.103	.087	.103

Hot.

1	.052	.049	.052	.045
2	.056	.049	.052	.045
3	.060	.051	.059	.047
4	.067	.052	.058	.050
5	.067	.055	.063	.048
6	.085	.063	.069	.050
7	.077	.055	.061	.045
8	.056	.045	.045	.057

Port Astern Turbine.

Cold.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
1	.071	.071	.068	.078
2	.072	.078	.079	.087
3	.103	.107	.096	.102
4	.103	.103	.099	.103
5	.102	.106	.099	.103

Hot.

1	.060	.050	.054	.057
2	.044	.045	.047	.052
3	.058	.070	.051	.066
4	.055	.053	.049	.051
5	.044	.034	.046	.030

Starboard Astern Turbine.

Cold.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
1	.067	.070	.066	.067
2	.080	.077	.065	.072
3	.103	.099	.090	.100
4	.103	.095	.086	.100
5	.103	.095	.082	.094

Hot.

1	.060	.047	.065	.044
2	.053	.047	.039	.040
3	.067	.060	.053	.060
4	.072	.049	.052	.050
5	.082	.035	.060	.030

Remarks.—On referring to the foregoing clearance data it will be noticed that the expansion of the turbines and blades is by no means uniform through, as for example at the position of the L.P. "wing" blades, that is, the 6th, 7th, and 8th expansion, where the blades are all of equal height, the expansion is unequal. Expansions 6, 7, and 8 of the starboard L.P. rotor all give a clearance of .103 at top when cold, but when heated up the respective clearances are then .085, .077, and .056, giving a variation of .085 − .056 = .029, that is $\frac{29''}{1000}$ difference in the clearance with equal blade heights. Again referring to the same L.P. cylinder, the clearances at the last three expansions are when cold, at top, .096, .097, and .087, which on being heated up alter to .060, .061, and .045, giving a difference of (.069 − .045) = .024, or $\frac{.024''}{1000}$ between the blades of the 6th and 8th expansion. Again, it will be noticed that the greatest clearance difference due to heating up is evident in the astern turbines, which show as much as $\frac{72}{1000}$ (.072) decrease in the clearance at the 5th expansion of the rotor (bottom) of the port astern turbine. Generally, the decrease in tip clearance due to heating increases from forward aft in any turbine, owing chiefly to the increase of blade heights being in that direction.

5. Type—Fast Passenger Steamer.

10,000 Horse-Power (approximate).

<table>
<tr><td></td><td colspan="6">No. 1.</td><td colspan="6">No. 2.</td><td colspan="6">No. 3.</td><td colspan="6">No. 4.</td></tr>
<tr><td>Boiler Pressure.</td><td colspan="6">180 lbs.</td><td colspan="6">183 lbs.</td><td colspan="6">180 lbs.</td><td colspan="6">180 lbs.</td></tr>
<tr><td>H.P. Initial Pressure.</td><td colspan="6">168 ,,</td><td colspan="6">165 ,,</td><td colspan="6">165 ,,</td><td colspan="6">165 ,,</td></tr>
<tr><td>H.P. Terminal Pressure.</td><td colspan="6">23 ,,</td><td colspan="6">22 ,,</td><td colspan="6">22½ ,,</td><td colspan="6">22½ ,,</td></tr>
<tr><td></td><td colspan="3">P.</td><td colspan="3">S.</td><td colspan="3">P.</td><td colspan="3">S.</td><td colspan="3">P.</td><td colspan="3">S.</td><td colspan="3">P.</td><td colspan="3">S.</td></tr>
<tr><td>L.P. Initial.</td><td colspan="3">22 lbs.</td><td colspan="3">22 lbs.</td><td colspan="3">21 lbs.</td><td colspan="3">21 lbs.</td><td colspan="3">21½ lbs.</td><td colspan="3">21½ lbs.</td><td colspan="3">22½ lbs.</td><td colspan="3">22½ lbs.</td></tr>
<tr><td>Condenser Vacuum.</td><td colspan="3">27 in.</td><td colspan="3">26¾ in.</td><td colspan="3">27 in.</td><td colspan="3">27 in.</td><td colspan="3">26¾ in.</td><td colspan="3">26½ in.</td><td colspan="3">27½ in.</td><td colspan="3">27 in.</td></tr>
<tr><td>Astern Turbines Vacuum.</td><td colspan="3">24¼ ,,</td><td colspan="3">24 ,,</td><td colspan="3">24½ ,,</td><td colspan="3">24½ ,,</td><td colspan="3">23½ ,,</td><td colspan="3">23½ ,,</td><td colspan="3">24½ ,,</td><td colspan="3">24¼ ,,</td></tr>
<tr><td></td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td></tr>
<tr><td>Revolutions per min.</td><td colspan="2">464</td><td colspan="2">484</td><td colspan="2">484</td><td colspan="2">451</td><td colspan="2">464</td><td colspan="2">463</td><td colspan="2">469</td><td colspan="2">490</td><td colspan="2">491</td><td colspan="2">458</td><td colspan="2">477</td><td colspan="2">479</td></tr>
<tr><td>Mean Revs. all Shafts.</td><td colspan="6">477</td><td colspan="6">459</td><td colspan="6">483</td><td colspan="6">471</td></tr>
<tr><td>Speed.</td><td colspan="6">22 knots (against tide).</td><td colspan="6">22.5 knots.</td><td colspan="6">23.3 knots.</td><td colspan="6">22.5 knots.</td></tr>
<tr><td></td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td><td colspan="2">H.P.</td><td colspan="2">P.</td><td colspan="2">S.</td></tr>
<tr><td>Dummy clearance.</td><td colspan="2">.028</td><td colspan="2">.028</td><td colspan="2">.020</td><td colspan="2">.030</td><td colspan="2">.028</td><td colspan="2">.020</td><td colspan="2">.030</td><td colspan="2">.029</td><td colspan="2">.021</td><td colspan="2">.03</td><td colspan="2">.028</td><td colspan="2">.020</td></tr>
</table>

NOTE.—Propellers, 6 ft. 6 in. diameter, 6 ft. pitch, mean slip, 18 per cent.

Gland Pressures.

H.P.		L.P.	
Steam.	Leak-off.	Steam.	Leak-off (shut).
2 lbs.	o	6 lbs.	8 in. vac.

NOTE.—Astern end gland of L.P. similar to ahead end gland.

In running "full astern" the reverse turbine initial pressure was 90 lbs., the L.P. ahead turbine $24\frac{1}{2}$ in. vacuum, the H.P. turbine 23 in. vacuum, the condenser $26\frac{1}{2}$ in. vacuum, and the mean revolutions about 400 per minute. The L.P. dummy clearance changed to .060.

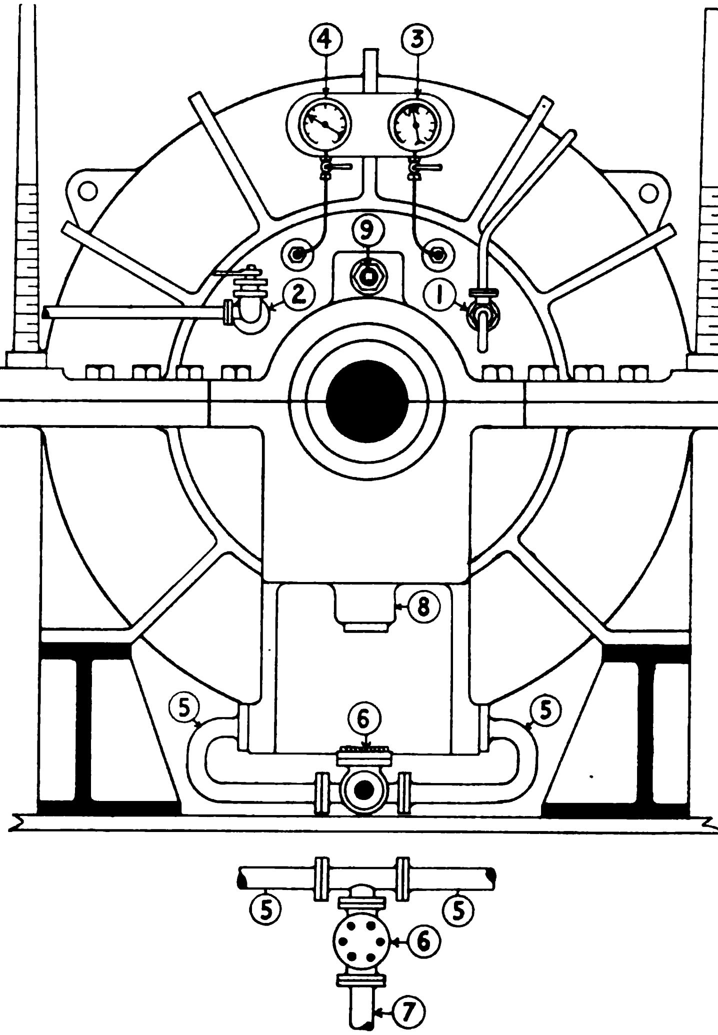
6. "**Carmania.**"—The following are a few particulars of this large turbine steamer built by Messrs John Brown & Co., Clydebank, to the order of the Cunard Line:—

Length - - - - - - -	678 feet.
Beam - - - - - - -	72 feet.
Boiler pressure - - - -	195 lbs. per sq. in.
No. of boilers - - - - - -	13
Revolutions - - - - - -	185 per minute.
Diameter of propeller - -	14 feet.
Pitch of propeller - - -	13 feet.
Equivalent horse-power - -	21,000
Speed - - - - - -	21 knots (trial).
Diameter of L.P. rotor - - -	138 in.

On the trial runs made on the Clyde the speed attained by the "Carmania" was 21.6 knots.

NOTE.—The pitch ratio of the propeller is equal to .9, and the area ratio probably about .6 or thereabout.

7. "**Express**" **Cunarders.**—The large Cunarders, "Lusitania" and "Mauretania," are fitted with four lines of shafting and four propellers, one to each length. The turbines are six in number, an H.P. on each outside shaft, and an L.P. and reverse turbine on each inside shaft. The reverse turbines are independent of the L.P. turbines, and are placed forward with the thrust block between the two (see sketch). The L.P. rotors are 188 in. in diameter without the blades, the largest height of blade used being 22 in. The line shafting is 22 in. in diameter, and the rotor spindles 33 in. diameter in the



End View (Aft) of L. P. Turbine, showing Steam Gland Connections and Drain Connections (original style of Gland).

1. Steam Inlet Cock admitting Steam to Outer Pocket.
2. Leak-off Cock on Inner Pocket connecting to 3rd or 4th L. P. Expansion.
3. Gauge on Outer Pocket.
4. Gauge on Inner Pocket.
5. Drains from Turbine Casing Bottom.
6. Light Non-return Valve.
7. Suction to "Wet" Air Pump.
8. Oil Drain to Oil Cooling Tanks from Thrust and Bearing.'

bearings, which are about 5 ft. in length. The boilers (cylindrical) are twenty-five in number, working at a pressure of 200 lbs. per sq. in. The speed attained has exceeded 26 knots. For complete data see *Engineering* of 2nd August 1907 and of 8th November 1907.

8. Record of Wear Down at Main Turbine Bearings.

NOTE.—The following figures refer to the wear down as measured by the "bridge gauge" (see page 210). The first column shows the actual gauge clearance when new, and the other column the subsequent increase of clearance due to wear down when tested one year later.

	Original Marking.	After One Year.
H.P. Forward	.033	.034 (tight)
H.P. Aft	.036	.035
Port L.P. Forward	.041	.044 (tight)
Port L.P. Aft	.04	.045
Starboard L.P. Forward	.037	.044
Starboard L.P. Aft	.032	.036

Tip Clearance.—To allow of rotor and blade expansion under heat a certain working clearance has to be allowed between the tips of the rotor blades and the inside surface of the casing, also between the tips of the casing blades and the outer surface of rotor drum. This of course works out as a certain per cent. of loss by leakage over blade tips, no useful work being done by the steam in passing through.

It is calculated that if the loss is 3 per cent. when the revolutions are 600 per minute, then the loss at 200 revolutions per minute will be 27 per cent., as the per cent. loss varies as the square of the revolutions, or, which is the same thing, as the square of the rotor diameter.

So that, allowing for the increased expansion due to larger drums, the loss increases by the square of the diameter, or in other words, by the area open to leakage.

The following typical examples of blade tip clearances and dummy clearances were taken from the turbines of a large channel steamer:—

9. BLADE TIP CLEARANCES.

Taken when cold.

H.P. Turbine.

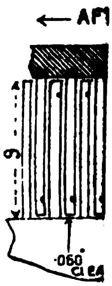
Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.041	.041	.052	.043	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{15}{64}$	$\frac{7}{32}$
„ 2	.051	.055	.059	.049	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
„ 3	.063	.055	.070	.061	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{1}{4}$
„ 4	.049	.045	.054	.051	$\frac{23}{64}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{11}{32}$

Starboard L.P. Turbine.

Rotor Drum, 68 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.070	.080	.068	.070	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{13}{64}$	$\frac{17}{64}$
„ 2	.072	.085	.070	.085	$\frac{5}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$
„ 3	.078	.090	.082	.092	$\frac{11}{32}$	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{17}{64}$
„ 4	.082	.092	.082	.085	$\frac{3}{8}$	$\frac{11}{32}$	$\frac{5}{16}$	$\frac{3}{8}$
„ 5	.085	.093	.085	.088	$\frac{7}{16}$	$\frac{11}{32}$	$\frac{11}{32}$	$\frac{11}{32}$
„ 6	.095	.123	.098	.115	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{7}{16}$
„ 7	.102	.125	.105	.112	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{15}{32}$
„ 8	.102	.115	.105	.113	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{1}{2}$



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CLEA

Port L.P. Turbine.

Rotor Drum, 68 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.069	.070	.075	.075	$\frac{15}{64}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{32}$
" 2	.072	.070	.088	.085	$\frac{17}{64}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{1}{4}$
" 3	.070	.075	.078	.088	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{16}$
" 4	.070	.078	.083	.092	$\frac{9}{32}$	$\frac{17}{64}$	$\frac{3}{8}$	$\frac{5}{16}$
" 5	.072	.073	.091	.090	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{5}{16}$
" 6	.102	.099	.122	.123	$\frac{13}{32}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$
" 7	.108	.108	.128	.115	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{27}{64}$	$\frac{7}{16}$
" 8	.117	.118	.132	.148	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{31}{64}$	$\frac{15}{32}$

Port Astern Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.085	.085	.067	.065	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{1}{2}$
" 2	.085	.088	.073	.073	$\frac{1}{8}$	$\frac{17}{32}$	$\frac{11}{64}$	$\frac{17}{32}$
" 3	.115	.113	.112	.098	$\frac{1}{8}$	$\frac{1}{2}$	$\frac{5}{32}$	$\frac{15}{32}$
" 4	.120	.115	.115	.103	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{7}{32}$	$\frac{1}{2}$

Starboard Astern Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.055	.062	.068	.070	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{17}{32}$
„ 2	.065	.064	.080	.082	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{7}{32}$	$\frac{17}{32}$
„ 3	.098	.102	.122	.128	$\frac{3}{16}$	$\frac{17}{32}$	$\frac{3}{16}$	$\frac{17}{32}$
„ 4	.098	.100	.125	.110	$\frac{7}{32}$	$\frac{17}{32}$	$\frac{7}{32}$	$\frac{17}{32}$

	Cold Setting.	On Trial.
H.P. Dummy Clearance -	$\frac{15}{1000}$	$\frac{30}{1000}$
S.L.P. „ „ - -	$\frac{36}{1000}$	$\frac{21}{1000}$
P.L.P. „ „ - -	$\frac{36}{1000}$	$\frac{27}{1000}$

It will be noted that the H.P. dummy clearance *decreased* when heated up, and the L.P.'s increased when heated up.

Observe that the reverse turbine longitudinal clearance is much more *aft* than forward, to allow of the heating up and expansion of the drum and casing, which takes effect on the after end of the turbine.

10. Blade Tip Clearance Top and Bottom.

TAKEN COLD.

H.P. Turbine.

Expansion.	Cylinder.		Rotor.	
	Top.	Bottom.	Top.	Bottom.
No. 1 - - - -	.040	.050	.032	.053
„ 2 - - - - -	.042	.056	.037	.057
„ 3 - - - - -	.052	.068	.041	.060
„ 4 - - - - -	.050	.061	.034	.057

P.L.P. Turbine.

Expansion.	Cylinder.		Rotor.	
	Top.	Bottom.	Top.	Bottom.
No. 1 - - - - -	.037	.067	.038	.083
„ 2 - - - - -	.037	.081	.041	.074
„ 3 - - - - -	.055	.078	.038	.076
„ 4 - - - - -	.057	.098	.038	.077
„ 5 - - - - -	.056	.114	.051	.111
„ 6 - - - - -	.072	.131	.076	.130
„ 7 - - - - -	.067	.131	.071	.131
„ 8 - - - - -	.072	.130	.070	.132

Turbine Pressure Data, &c.

11. The following results obtained in trial runs of the Isle of Man passenger steamer, "Manxman," give a good idea of the figures usually obtained in turbine steamers of this class:—

Mean speed of two runs - - - - - - -	23.14 knots.
Boiler pressure per square inch - - - - - -	192 lbs.
Steam in high-pressure turbine - - - - - -	180 „
„ low-pressure turbine, port - - - - -	20 „
„ „ „ starboard - - - -	20 „
Vacuum in condenser, port - - - - - -	28.25 in.
„ „ starboard - - - - -	28.4 „
Revolutions per minute, high-pressure turbine - -	533
„ „ low-pressure „ - - - -	609
Temperature of feed-water leaving heater - - -	180° Fahr.
Air-pressure in stokehold - - - - - - -	1.5 in.

12. The following data refer to a cross-channel steamer:—

Full Speed Ahead (all Turbines).

Speed	22 knots.
I.H.P.	6,500.
Boiler	160 lbs.
H.P. turbine	140 "
L.P. port	20 "
L.P. starboard	20 "
L.P. port (astern)	23 in. vacuum.
L.P. starboard (astern)	23 " "
Condenser vacuum	$24\frac{1}{2}$ " "
Revolutions	630 per minute (all shafts).
Propeller pitch	4 ft. 6 in.
Slip	21 per cent.

13. Gauge Indications.—The following figures from actual practice give a fair idea as to the usual pressures and vacuum carried in the turbines of a steamer of moderate power:—

Type—Channel Steamer.

Boiler pressure	160 lbs.
H.P. turbine (initial)	150 "
L.P. ahead turbines (initial)	24 "
Astern turbines of L.P. shafts	25 in. vacuum.
Condenser	27 " "
Revolutions	650 (all shafts).

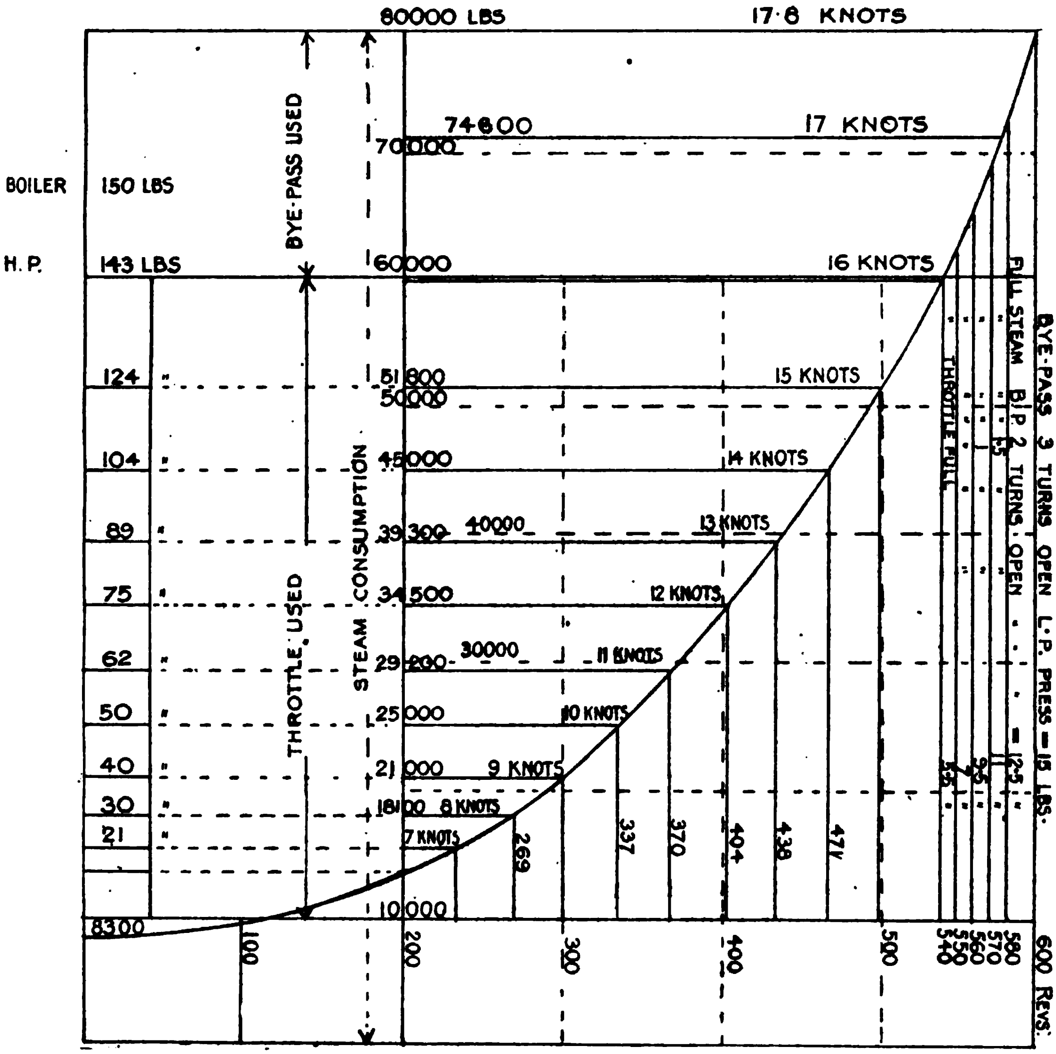
NOTE.—When running ahead at reduced speed with *all turbines* working, the H.P. pressure is 80 lbs., L.P. pressures 15 lbs., and condenser 28 in. vacuum.

Type—Channel Steamer.

(14a.) Running Full Power Ahead with all Three Turbines.

Boiler pressure	155 lbs. (gauge).	
H.P. turbine pressure	146 " "	
Port L.P. turbine pressure	13 " "	
Starboard L.P. turbine pressure	$12\frac{1}{2}$ " "	
Condenser vacuum	27 in.	
H.P. forward steam gland pressure	1 lb.	These steam glands have no "leak-off" connections.
H.P. aft steam gland pressure	1 "	
L.P. forward steam gland pressures	2 lbs.	
L.P. aft steam gland pressures	$3\frac{1}{2}$ "	
Coal per shaft horse-power per hour	1.8 "	

NOTE.—Observe that the L.P. turbine steam gland pressures are *higher* than the H.P. turbine gland pressures, this being required to prevent the admission of air into the turbine casing, which would destroy the L.P. vacuum.



Steam Consumption, Revolutions, and Speed Curve.

The above curve is developed from the steam consumption, revolutions, and speed of a turbine coastal steamer of about 5,000 horse power.

[*To face page* 170.

(14b.) Running Ahead with Two Outside (L.P.) Turbines only.

(Centre shaft running idly in vacuum at about $\frac{1}{3}$ revolutions of outside shafts.)

Speed	15 knots.
H.P. turbine pressure	20 in. vacuum.
L.P. turbine pressures, port and starboard	20 lbs. (gauge).
Condenser vacuum	$27\frac{1}{2}$ in.

(14c.) Running Full Speed Astern with Two Outside (L.P.) Turbines.

Speed	14 knots.
H.P. turbine pressure	20 in. vacuum.
L.P. turbine pressures, port and starboard	80 lbs. (gauge).
Condenser vacuum	27 in.
Revolutions	500.

NOTE.—When running ahead with all turbines one inch increase of vacuum gave 23 extra revolutions to each L.P. turbine shaft. The importance of a high degree of vacuum will thus be obvious.

In the steamer referred to above, the pitch of the outside propellers is 4 ft. 3 in., and of the H.P. (centre) 4 ft. 6 in. This difference of pitch is accounted for by the fact that the centre or H.P. shaft runs at less revolutions than the outside shafts, and therefore necessitates a larger pitch to obtain the same screw and ship advance. An example will make this clear.

Speed	20 knots.
Slip	28 per cent.
L.P. turbine revolutions	662 per min.
H.P. turbine revolutions	620 ,,

To find the required pitch of propellers :—

$$\text{Knots} \times 6080 = \text{Pitch} \times \text{Revolutions} \times 60 \times \text{Effective \% advance.}$$

$$\text{Therefore, } 20 \times 6080 = \text{Pitch} \times 662 \times 60 \times .72,$$

$$\text{and, } \frac{20 \times 6080}{662 \times 60 \times .72} = 4.2 \text{ feet pitch (outside propellers),}$$

$$\text{and, } \frac{20 \times 6080}{620 \times 60 \times .72} = 4.5 \text{ feet pitch (centre propeller).}$$

NOTE.—$100 - 28 = 72$, and $\frac{72}{100} = .72$ effective advance of screw.

15. Consumption.—The consumption for this steamer worked out as 1.8 lb. of coal per *shaft horse-power* per hour, the power being calculated by means of the Denny & Johnson torsion meter (see page 262). This consumption, if reduced to the corresponding indicated horse-power, would be less per horse-power in proportion, as the shaft or transmitted horse-power is obviously less than what the indicated horse-power would be, were it possible to indicate the power.

The slip at full ahead speed worked out as 30 per cent. which, at an ahead speed of from 10 to 14 knots, fell off to 11 per cent.

"Steam Gland" Connections.—Sometimes this cock, fitted to the end of the turbine casing at the "steam gland" port, or "outer pocket," is of the two-way type, and is arranged either to (1) give direct steam to the gland, or (2) connect the gland with the condenser. This is necessary in the case of the H.P. steam glands, as, when the two outside shafts are working only, the H.P. turbine is then merely revolving inertly in a vacuum, due to the water action on the screw. Under these conditions the cocks can be opened to the condenser, but when the H.P. is running ahead with full steam, the cocks can then be opened to admit a steam pressure to the glands.

NOTE.—With the above arrangement the "leak-off" port and pipe connection (see page 123) are both omitted.

Vacuum in Idle Turbines.—When running ahead with all three turbines the reverse turbines are revolving in a vacuum of from 22 to 25 in., and when running ahead or reverse with the outside shafts only, the centre or H.P. turbine is then revolving in a vacuum also of about 25 in. In this case the centre propeller is caused to revolve by the action of the water on the blades, being forced round in the ahead direction if the outside shafts are running ahead, and forced round in the astern direction if the outside shafts are running astern at the time.

Importance of High Vacuum.—As before mentioned, a high intensity of vacuum is of considerable importance to a turbine, as the total expansions of steam may be taken to vary in proportion to the amount of vacuum obtained.

$$\text{Suppose} \begin{cases} \text{Initial pressure in H.P. turbine} = 140 \text{ lbs. gauge,} \\ \text{Vacuum in condenser} = 26 \text{ in.,} \end{cases}$$

the actual vacuum at the last L.P. turbine row will probably be somewhat less, say 24 in., so that

$$24 \text{ in.} \div 2 = 12 \text{ lbs.,}$$

$$\text{and } 15 - 12 = 3 \text{ lbs. absolute terminal pressure.}$$

$$\text{Therefore, } (140 + 15) \div 3 = 51.6 \text{ expansions of steam in all (by pressures).}$$

If therefore the vacuum is increased to say 28 in. in condenser and 26 in. at L.P. turbine exhaust the result will be as follows:—

$$26 \text{ in.} \div 2 = 13 \text{ lbs.,}$$

$$\text{and } 15 - 13 = 2 \text{ lbs. absolute terminal pressure.}$$

$$\text{Therefore, } (140 + 15) \div 2 = 77.5 \text{ expansions of steam in all (by pressures).}$$

From the foregoing it will be evident that the economy on turbine engines depends greatly on the vacuum carried.

16. Type—Channel Steamer.

(9,000 Horse-Power.)

Speed Trials (Ahead).

Revolutions per Minute (mean of three shafts).	H.P. Turbine Pressure.	Port L.P. Ahead Turbine Pressure.	Starboard L.P. Ahead Turbine Pressure.	Speed in Knots.
290	10 lbs.	21 in. vacuum	21 in. vacuum	10.68
409	35 „	12 „ „	12 „ „	14.73
512	60 „	0	0	18.12
608	115 „	22 lbs.	22 lbs.	20.82
680	152 „	23 „	23 „	22

17. Astern Trials (Two L.P. Turbines only).

H.P. Turbine Pressure.	Port L.P. Turbine Pressure.	Starboard L.P. Turbine Pressure.
18 in. vacuum	82 lbs.	80 lbs.
25 „ „	50 „	50 „

NOTE.—Average revolutions 540 per minute, and mean astern speed 14.4 knots, at higher L.P. pressure shown above.

18. Turbines in Vacuum.—(1) When running ahead with all three turbines the reverse (L.P.) turbines are revolving inertly in a vacuum.

(2) When running ahead with the two outside turbines only, the H.P. turbine and the two reverse (L.P.) turbines are revolving in a vacuum.

(3) When running astern with the two outside turbines, the two L.P. ahead turbines and the H.P. turbine are revolving in a vacuum.

The average vacuum indications in a case noted by the writer were as follows:—

In No. (1), reverse turbine vacuum, 24 to 26 in.

„ (2), H.P. vacuum, 25 to 27 in.

„ (3), L.P. ahead turbines, 26 to 27½ in.

19. Steam Gland Pressures.—With the steam gland and least off arrangements as formerly fitted at each end of the turbines, the pressures indicated, when running ahead full power, were as follows:—

	Outer Pocket (Steam Inlet).	Inner Pocket (Leak-off to L.P. 3rd Expansion).
H.P. Turbine (forward and aft glands) -	1 lb.	10 to 15 in. vacuum
L.P. Turbine (forward and aft glands) -	1 „	5 to 15 „ „

The "steam to glands" pipe is led off the main steam pipe, and is reduced in pressure by throttling the cock or valve to about 40 lbs.; this is, of course, still further reduced at the "outer pocket" of the glands by wire-drawing at the cock on the gland itself.

The inner pocket of the steam glands leaks off the small quantity of steam which finds its way past the brass rings and collars to the 3rd expansion of the L.P. turbines. All the leak-off connections from all three turbines forward and aft are connected up in this way when running ahead with all three turbines. When running ahead or astern with the two outer (L.P.) turbines only, the H.P. leak-off cocks are changed over direct to the condenser with the pressure results shown below.

20. Running Ahead with Two L.P. Turbines only.

	Outer Pocket (open to Condenser).	Inner Pocket (Leak-off to L.P. 3rd Expansion).
H.P. Turbine (forward and aft glands) -	27 in. vacuum	5 lbs. steam pressure

NOTE.—If the speed is reduced when running ahead with all three turbines, the "outer pocket" steam inlet pressure increases by a few pounds, and the "inner pocket" leak-off vacuum also increases by a few inches.

It should also be noted that the H.P. steam gland cocks are of the "two-way" type, to allow of the change over to condenser when required; whereas the L.P. steam gland cocks are only "one-way" type. The "leak-off" cocks from all turbines are single-way cocks.

21. Turbine Dimensions.—The following are the principal turbine dimensions of a large cross-channel steamer of about 10,000

I.H.P., speed 22 knots, and revolutions about 500, the propeller pitch being 5 ft. 6 in.:—

	H.P. Rotor.	L.P. Rotor.	Reverse Rotor.
Diameter	48 in.	68 in.	48 in.
Length	68 „	90 „	58 „
Number of Expansions	4	8	4
Number of Blade Rows in each Expansion	12	6	12
Total number of Rows of Blades	48	48	48
Blade heights in each Expansion—			
1st Expansion	$1\frac{3}{8}$ in.	$1\frac{3}{8}$ in.	$\frac{3}{4}$ in.
2nd „	2 „	2 „	$1\frac{1}{2}$ „
3rd „	$2\frac{3}{4}$ „	$2\frac{3}{4}$ „	3 „
4th „	4 „	4 „	3 „
5th „	...	$5\frac{1}{2}$ „	...
6th „	...	8 „	...
7th „	...	8 „	...
8th „	...	8 „	...

NOTE.—It should be noted that the casings will contain the same number of blade rows as the respective rotors, that is, 48 rows in each.

Coal Consumption.—Regarding the all-important question of coal consumption, results have proved that at low or moderate speeds the reciprocating engine burns less per I.H.P. per hour than the turbine, but at high or maximum speeds the reverse is the case, the turbine showing in some cases an economy of as much as 20 per cent. and more over the reciprocating engine.

At the present time deep-sea turbine steamers compare very favourably with the ordinary reciprocating engine in the matter of coal consumption, and for river or channel service the turbine steamers often run at a lower consumption than those of the reciprocating type. Cruisers of the "Indomitable" class show the remarkably low consumption at top speed of about 1.3 lbs. of coal per horse-power hour, and at reduced cruising speeds and powers of from 1.8 to 2.2 lbs. per horse-power hour. These results are much below reciprocating engine practice for similar vessels.

The most conclusive tests as showing the superiority of the turbine over the triple engine at high speeds were those carried out some time ago by order of the Admiralty in the sister vessels "Topaze," "Sapphire," "Diamond," and "Amethyst," which were all designed and constructed similar in every particular, with the difference that the "Topaze," "Sapphire," and "Diamond" were supplied with triple-expansion engines, and the "Amethyst" with turbines.

The displacement of each vessel was 3,000 tons, and the estimated I.H.P. required for a speed of 21.7 knots was 9,800; it is therefore only reasonable and correct to assume that the power required to drive each vessel would be equal for the same speed. Working on this basis the results shown indicate a decided advantage in coal consumption in the turbine-propelled "Amethyst," as compared with the triple-expansion engined "Topaze" at the higher speed, but the reverse at lower speeds. The clear gain in coal at the maximum speeds is quite remarkable and constitutes a strong argument in favour of turbines; at 14 knots the conditions are, so far as economy is concerned, more equal; but when the speed was increased to 18 knots, it was found that the consumption on board the "Amethyst" was something like 20 per cent. less; at 20 knots it was nearly 30 per cent. less; and at the higher speed the improvement was still greater. The influence of this economy on the radius of action is very marked; for instance, the turbine-propelled ship could, with her 750 tons of coal on board, steam 3,160 sea-miles at 20 knots, as compared with 2,140 miles by the cruisers fitted with the ordinary machinery.

22. Coal Consumption.—A few examples of coal consumption, noted by the writer from actual practice, are appended.

(A) I.H.P. 8,500, speed 22 knots, consumption about 6 tons per hour.

$$\text{Then, } \frac{6 \times 2240}{8500} = 1.58 \text{ lb. per hour per I.H.P.}$$

(B) Shaft horse-power (not I.H.P.) 6,500, speed 20.7 knots, consumption about 5.2 tons per hour.

$$\text{Then, } \frac{5.2 \times 2240}{6500} = 1.79 \text{ lb. per hour per shaft H.P.}$$

$$\text{and, } 1.79 \times .9 = 1.61 \text{ lbs. per I.H.P. per hour.}$$

(C)

I.H.P. - - - -	9,000
Speed - - - - - - -	22.8 knots.
Mean revolutions - - - -	520 per minute.
Boiler steam - - - - -	175 lbs.
H.P. turbine - - - - -	140 ,,
L.P. turbines - - - - -	12 ,,
Reverse turbines - - - -	22 in. vacuum.
Condenser - - - - - -	$27\frac{1}{2}$,, ,,
Consumption - - - - -	6.67 tons per hour.

$$\text{Coal per I.H.P. per hour} = \frac{6.67 \times 2240}{9006} = 1.66 \text{ lb.}$$

See also pages No. 152, 155, 158, 171, 194, 195, 203, 205, 206, 207, 208.

Running Astern.—In running astern, the consumption increases above that required for ahead, as the astern turbines are naturally not so efficient as those designed for ahead work only.

23. Blade Dimensions.—The following dimensions, taken from the L.P. turbines of a large ocean-going steamer of about 12,000 I.H.P., will afford a fair idea of the proportions between the blade widths, blade heights, and blade tip clearances adopted in actual practice. The L.P. turbines consist of 8 expansions arranged as follows:—

L.P. Turbines.

Rotor Drum, 7 ft. 9 in. Diameter.

Expansion.	Blades.	Blade Tip Clearances (Cold).
1st	10 Rows of $\frac{3}{8}''$ blades $1\frac{3}{8}''$ high and $1\frac{1}{8}''$ pitch - -	.08″
2nd	10 " $\frac{3}{8}''$ " $2''$ " $1\frac{1}{4}''$ " - -	.08″
3rd	10 " $\frac{3}{8}''$ " $2\frac{3}{4}''$ " $1\frac{3}{8}''$ " - -	.08″
4th	10 " $\frac{3}{8}''$ " $4''$ " $1\frac{1}{2}''$ " - -	.09″
5th	10 " $\frac{1}{2}''$ " $5\frac{1}{2}''$ " $1\frac{7}{8}''$ " - -	.10″
6th	10 " $\frac{1}{2}''$ " $7''$ " $2''$ " - -	.12″
7th	10 " $\frac{1}{2}''$ " $7''$ " $2''$ " - -	.12″
8th	10 " $\frac{1}{2}''$ " $7''$ " $2''$ " - -	.12″

Dummy clearance, .03″ (or $\frac{30}{1000}''$).

Observe that the blades of the 6th, 7th, and 8th expansions are all of the same height and pitch, the only difference being that of blade section, the 7th and 8th expansions having blades of a flatter surface section and a greater circumferential pitch, as the packing pieces are thicker.

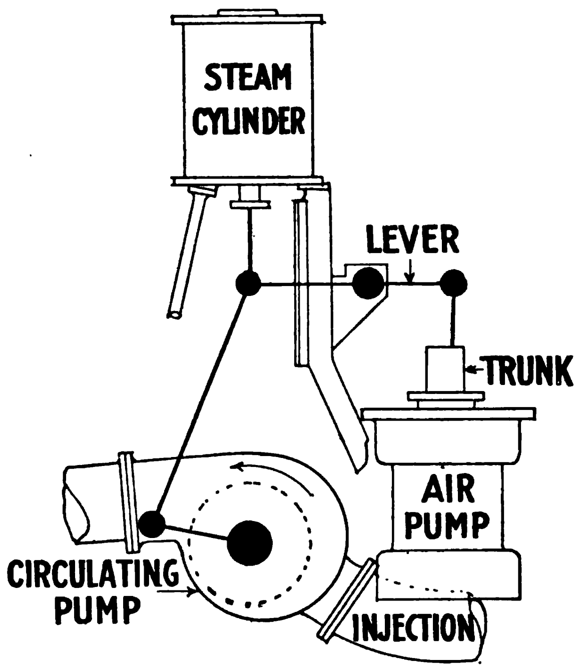
In steamers of from 5,000 to 8,000 estimated I.H.P. the blades vary in width from about $\frac{5}{16}$ in. in the first H.P. expansion to $\frac{1}{2}$ in. in the L.P. expansions, and in height from $1\frac{1}{16}$ in. in the H.P. turbine to about 8 in. or 10 in. in the L.P. turbines, depending on the diameter of rotor drum and the number of revolutions. In general, the larger the rotor drum diameter the shorter the blade heights.

Pumps.—All pumps fitted are of the independent double-acting type, and are as follows:—

Air Pumps.—The most recent practice consists of the fitting of independent twin air pumps of the Weir type, technically known as "wet" air pumps. These pumps draw as usual from the bottom of the condenser, and deliver the water into the feed tank and gravity

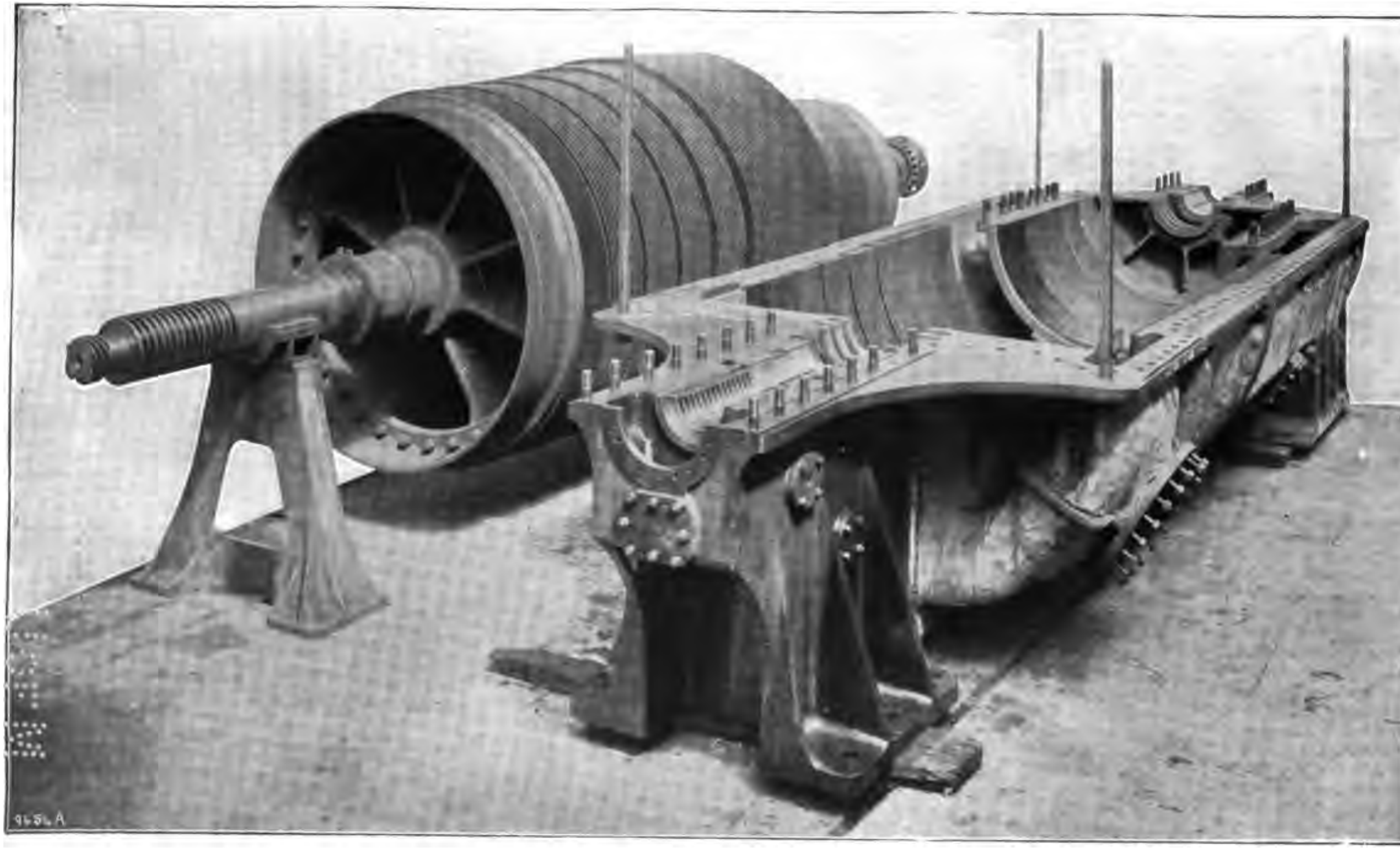
feed filter connections; from these the water is pumped by other independent donkey pumps into the surface feed heater and float tanks overhead, and from there the water is taken by the feed pumps proper and delivered direct into the boilers, purified and heated.

"**Dry**" **Air Pump.**—To obtain maximum efficiency in turbines, a high vacuum is of the utmost importance, and to produce this a special pump, known as a "dry air pump," has been recently introduced by Messrs G. & J. Weir, in addition to the ordinary independent air pumps as formerly fitted.



Air and Circulating Pumps.

The Weir "dry air pump" draws the air or vapour only from the condenser, and the ordinary air pump draws away the vapour still left and the condensed water. The dry air pump is therefore placed high up in position (usually above the circulating pump engine) with the suction branch from the condenser and the discharge pipe led away over the ship's side. By the aid of this special pump the vacuum carried has been as high as 29 inches. The dry air pump chamber is kept cool by means of a water jacket. Regarding the subject of vacuum the Hon. A. C. Parsons says: "An addition of 1 in. to the vacuum over 26 in. deducted 4 per cent. in the condenser from the steam used; a further increase of the same amount, 1 in.,



L.P. Ahead and Reverse Turbines.

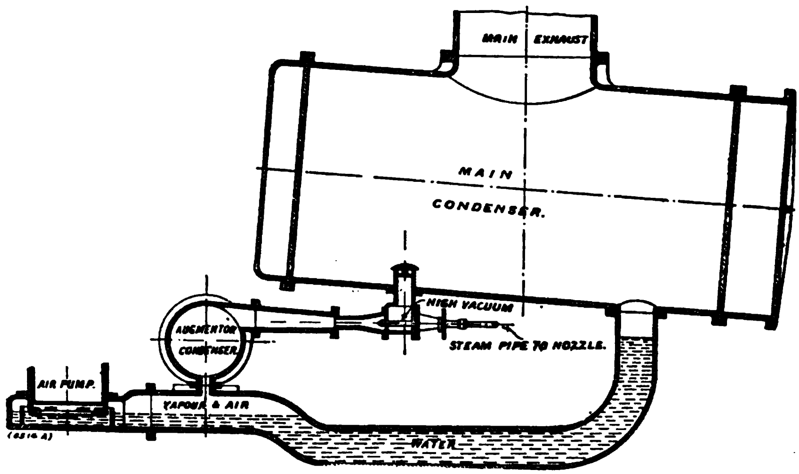
(Khedive's Yacht "Mahroussa.")

Messrs A. & J. Inglis Ltd.

[*To face page* 178.

meant a further gain of $4\frac{1}{2}$ per cent., while 29 in. brought the steam consumption down $5\frac{1}{2}$ per cent. more."

Another arrangement, called a "vacuum augmenter," devised by the Parsons Company, has been applied to increase the vacuum above that obtained by the air pump, and, as shown in the sketch, consists of a smaller auxiliary condenser placed below the main condenser and connected to it by a pipe having a conical contracted portion through which a jet of steam is forced. The effect of this is to exhaust most of the air and vapour from the condenser, and deliver it to the air pump. By arranging the air pump suction pipe with a dip as shown, the air and vapour is prevented from returning to the condenser by the water contained in the pipe, thus forming what may be termed a "water-seal." The "vacuum augmenter" has been found



Vacuum Augmenter.

in some cases to increase the vacuum by about 2 inches above that obtained with the air pump.

It should, however, be stated that the "Weir" dry air pump has been fitted in most of the larger turbine steamers.

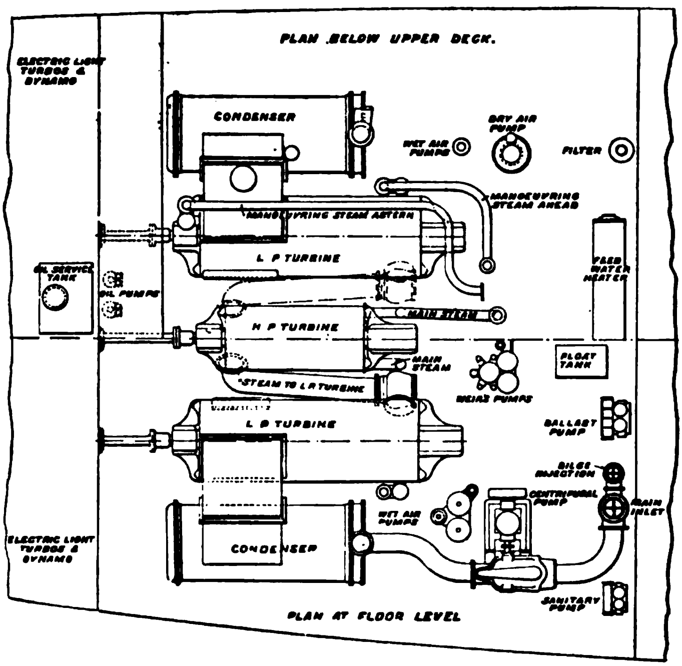
Circulating Pumps.—These two pumps, one for each condenser, force the cold sea water through the condenser tubes as in ordinary practice, and assist, in combination with the air pump, in producing condensation of the steam and the formation of a vacuum.

Feed Pumps.—As before stated, these pumps, arranged in pairs, are usually of the "Weir" or some other well-known patent type.

Various Pumps.—Other sets of pumps are required to circulate the cooling water through the bearings and cooling tanks, to force the oil at a pressure of 8 or 10 lbs. per square inch through the bearings, and for numerous other services, such as bilges, ballast tanks, &c. It

will thus be apparent that the number of independent service pumps required is in excess of those commonly fitted in engine-rooms of the reciprocating type.

Revolutions.—As previously stated, high revolution speed is necessary for the economical running of turbine engines. In the Clyde river turbine steamers the L.P. shafts revolve at about 800 revolutions per minute, and the H.P. shaft from 550 to 650 revolutions per minute, but the revolutions vary considerably with the type of vessel and speed desired. In ocean-going steamers the revolutions are much less than the above, the turbines of the "Lusitania" revolving at only 180 per minute. This means a corresponding increase in the

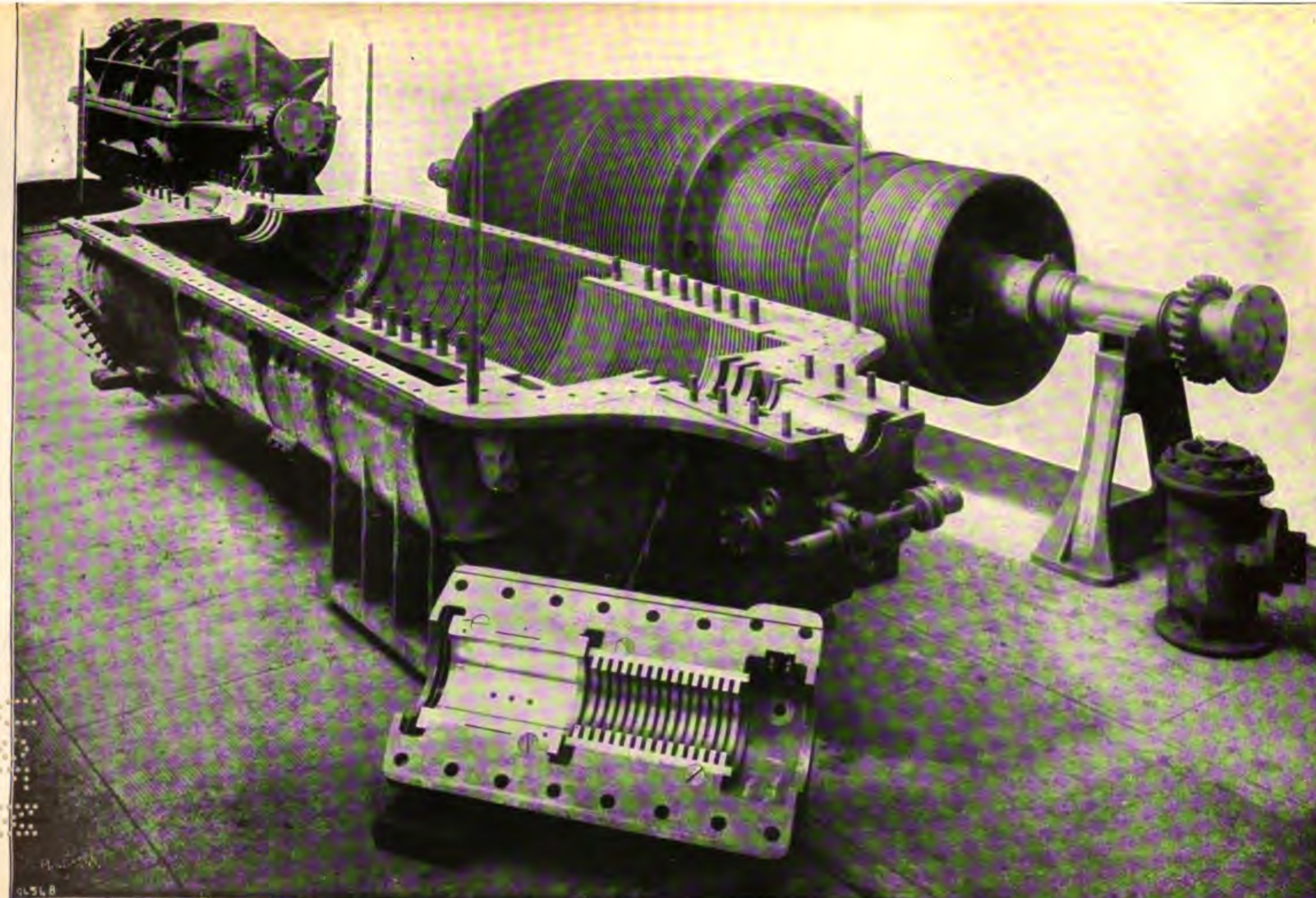


Plan of Turbine Room.

size of the rotor, and therefore increased weight of machinery. Recently the speed of all three shafts has been arranged to run at very nearly the same number of revolutions per minute. In the three large Irish Channel steamers, "St George," "St David," and "St Patrick," recently completed for the Great Western Railway Company, the revolutions run were just under 500 per minute at full power (10,000 I.H.P.), for a trial trip speed of 23 knots.

Governors.—Governors are now rarely fitted to turbines, but the "Carmania," it may be stated, is supplied with "Aspinall" type governors.

Speed Indicator.—The revolutions are indicated by an appliance



L.P. AHEAD AND REVERSE TURBINES.

Messrs A. & J. Inglis Ltd. The H.P. Turbine shown in background. (Khedive's Yacht "Mahroussa.")

[To face page 180.

known as a "Speed Indicator," or "Tachometer," which is fitted with a figured dial and pointer, and is specially employed in registering high revolution speeds in dynamos, &c. This instrument registers 1 revolution in 10.

Reversing.—Contrary to the first impressions and general expectations, turbine steamers have proved that quick stopping and reversing can easily be accomplished if the reverse turbines are made of sufficient power.

It should be stated, however, that sudden reversing is severe on the turbine, as excessive vibrations are set up by the change of propeller rotation.

Corrosion, Wear, &c.

Wear.—The wear on the brass blades of the turbines is practically *nil* after some years of service, the friction of the steam apparently having little or no effect on them. Wear on the bearings is very slight indeed, and only shows when something gets out of line. It is understood that when running the turbines balance themselves all round, or in other words are floating more or less, thus minimising the wear to a great extent.

As mentioned previously, very little pressure bears on the thrust block, as the blades and dummy pistons together take up and counterbalance most of the propeller thrust.

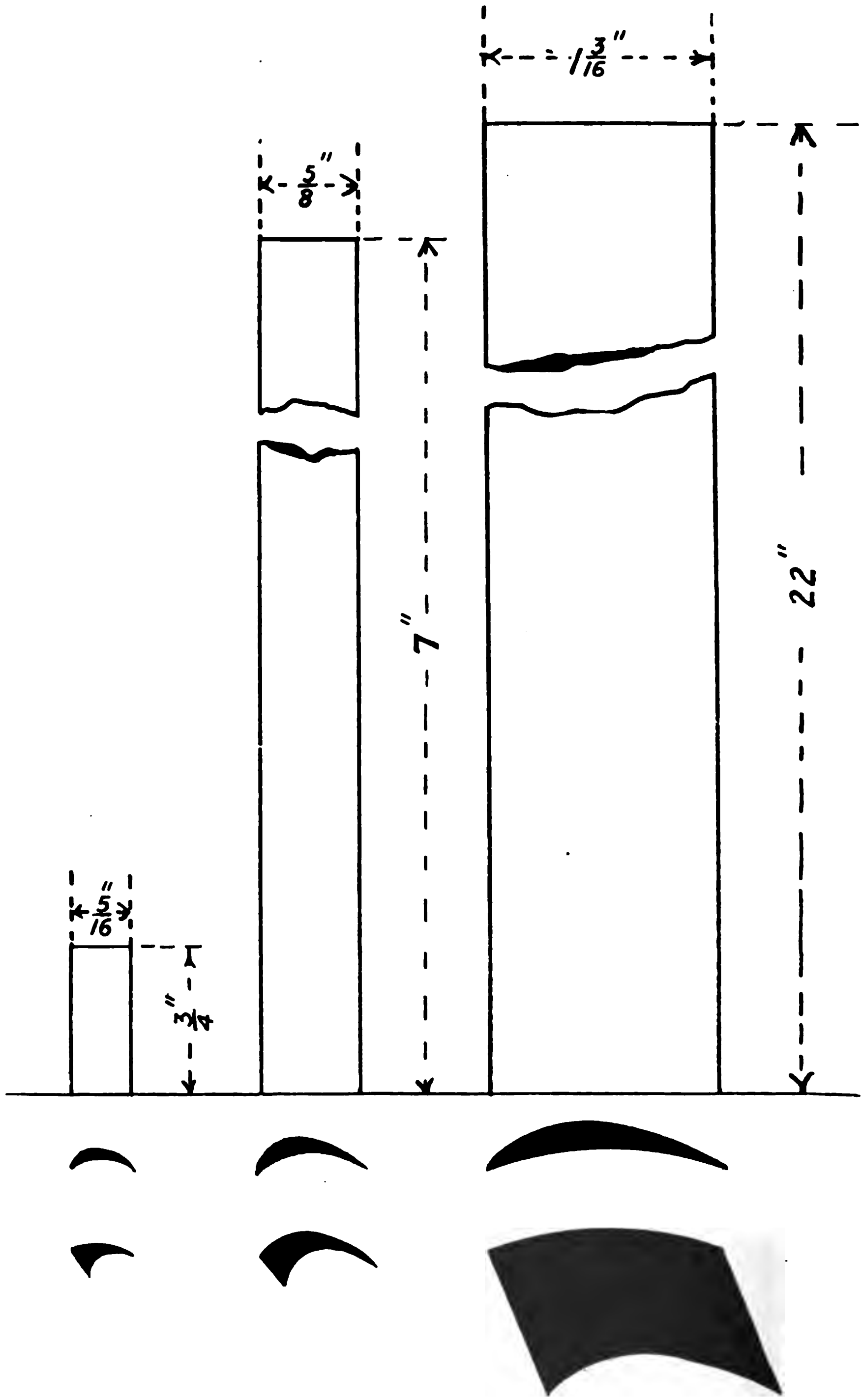
It is also evident that gyrostatic action of the rotors is so slight as to be negligible altogether. "Whipping" is also eliminated owing to the very careful balancing to which the various parts are subjected while under construction.

Wear of Gland Rings.—The loose brass rings which constitute the steam gland packing occasionally show decided signs of wear after several months' service, and if not renewed some of the rings are apt to give way altogether. A broken ring might thus result in damage to the blades, as the broken pieces might pass into the rotor drum and from there out again into the case and among the rings of blades. This is quite possible, as the wheels or centre have eight large openings cast in them which open up the drum at each end to the exhaust pressure of the turbine, and therefore to the casing.

If one or two of the H.P. turbine gland rings wore thin and finally broke, the pieces might pass into the drum as described, and then out of the drum and into the L.P. turbine with the exhaust steam, and damage to the L.P. blades would be likely to follow.

From the foregoing it will be obvious that the gland rings should be regularly renewed and kept up to their proper size and strength.

Wear of Thrust Block.—After long service the thrust block rings and collars show practically no wear, the surfaces being merely polished, and in many cases the original tool marks can still be made out. The wear, measured by the writer, in one case was only $\frac{2}{1000}$ of an inch after two years' hard running.



Types of Blades and Packing Pieces (Full Size).

The first is from the 1st H.P. Expansion of a large Channel Steamer, the second is from the last L.P. Expansion of a large Ocean-going Steamer, and the third is from the last L.P. Expansion of the "Express" Cunard Steamers "Lusitania" and "Mauretania."

Low-Pressure Turbine Casing (Upper Half), R.M.S. "Viper."

View showing Gland Steam Inlet and Outlet, Ahead Dummy Casing, Ahead and Reverse Expansions.

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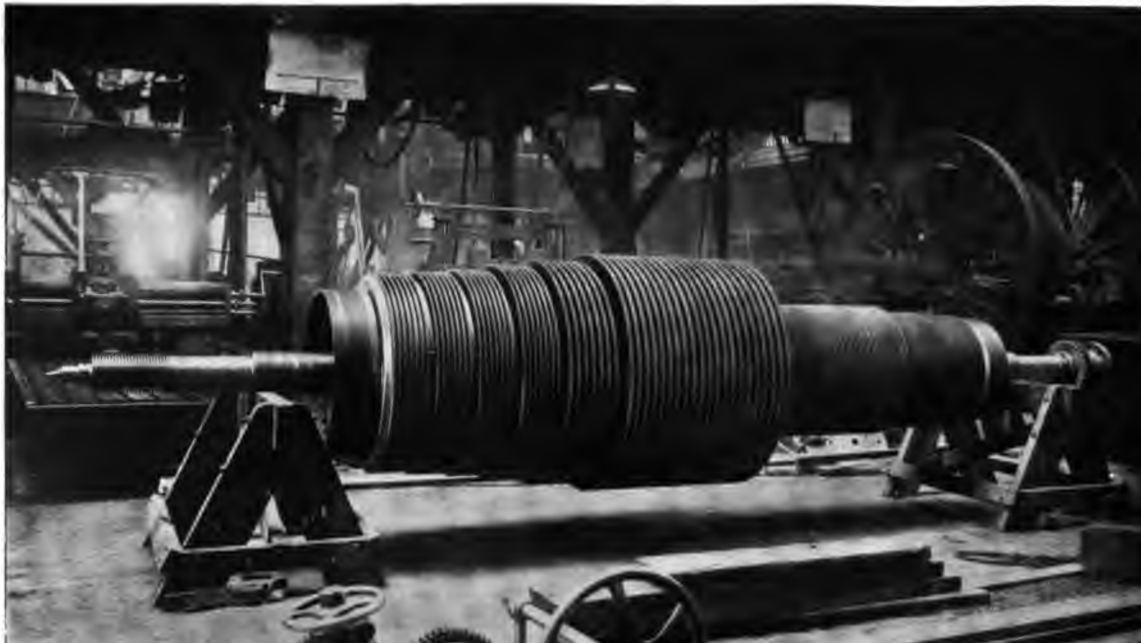

Owners—Messrs G. & J. Burns Ltd. *Builders*—The Fairfield Shipbuilding & Engineering Co. Ltd.

Low-Pressure Rotor, with Reverse Turbine Complete, R.M.S. "Viper."

View showing Thrust Collars, Main Bearings, Gland Collars, Ahead Dummy Piston, Ahead Turbine Expansions, Reverse Turbine Expansions, and Reverse Dummy Piston.

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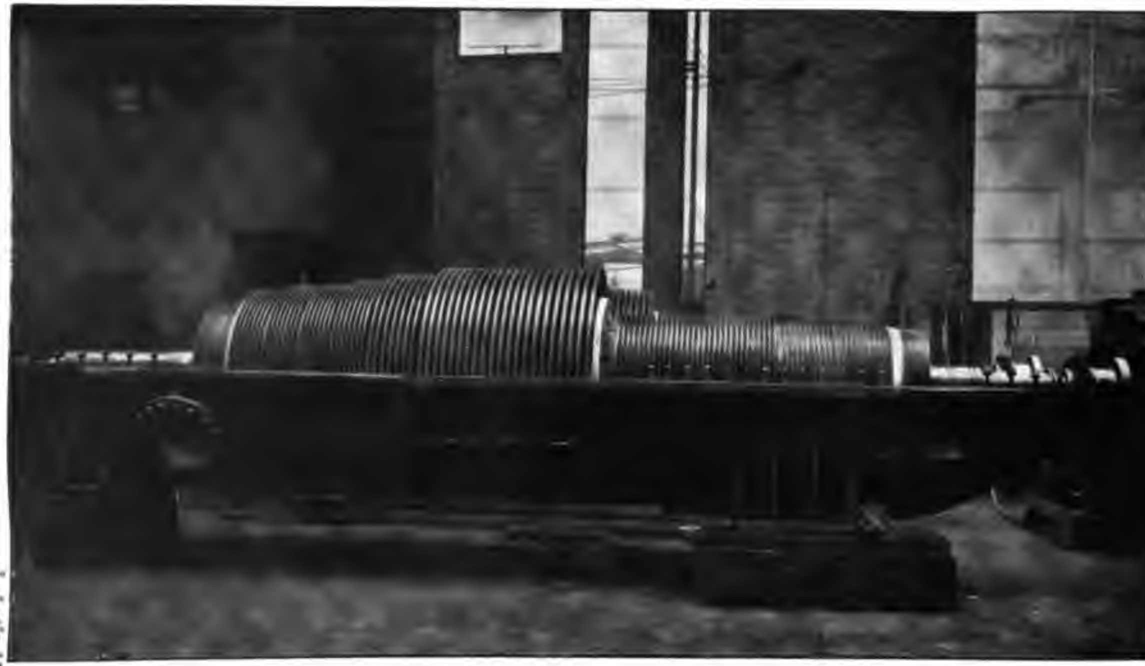

Low-Pressure Rotor, in Lower Half Casing, R.M.S. "Viper."

View showing Ahead Steam Admission, Ahead Dummy Piston, Complete Ahead and Reverse Expansions, also Reverse Dummy Piston.

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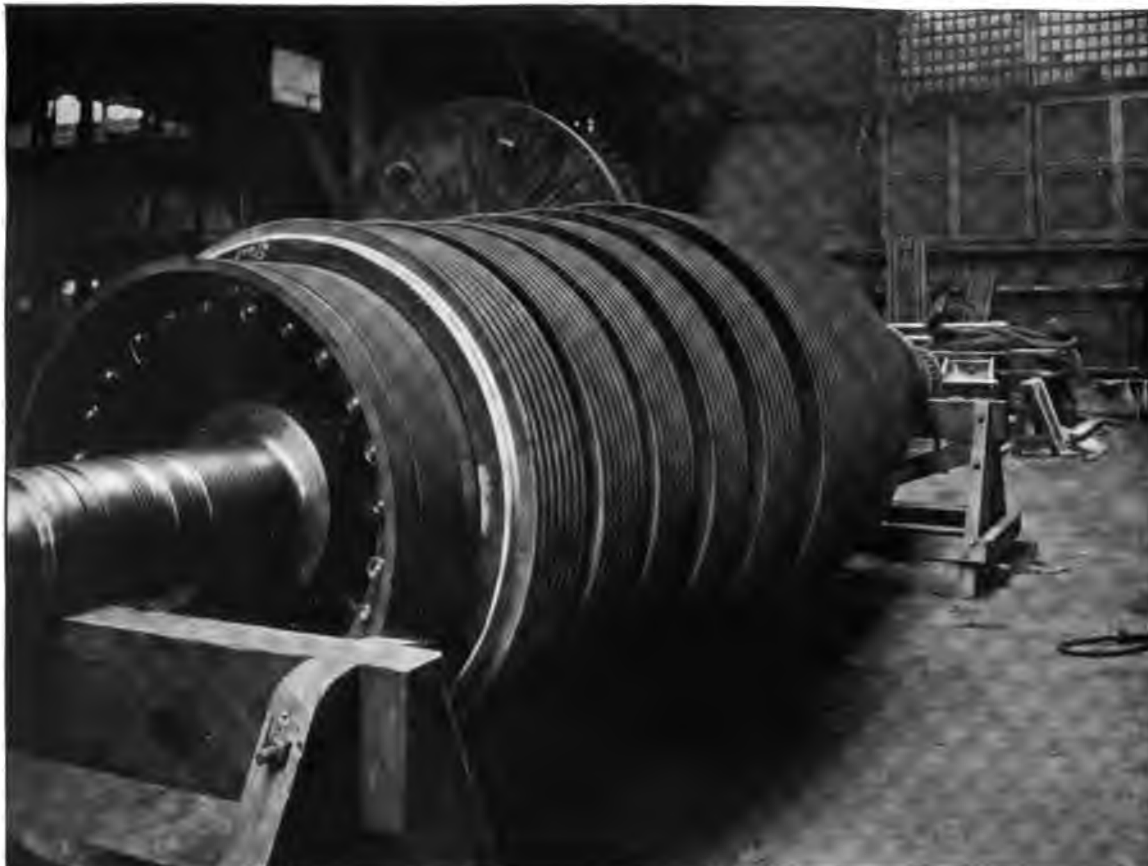


Low-Pressure Rotor Complete, R.M.S. "Viper."

View showing Gland Collars, Ahead Dummy Piston, and Steam Expansions.

To face page 182.

Wear of Bearings.—The wear of the bearings is also very slight indeed, and in some cases is a negligible quantity, being merely $\frac{2}{1000}$ or $\frac{3}{1000}$ either *down or up*, as sometimes the bearings appear to wear about equally all round, which circumstance would indicate that the rotor when running is in that condition known as "floating." The wear down of the L.P. rotors in a case tested by the writer was only $\frac{4}{1000}$ of an inch after two years' service.

Wear of Blades.—As a general rule the turbine blades after fairly long service show no sign of wear whatever, the only difference noticed being the darkened colour produced by the effect of the heat of the steam, otherwise the blades appear unchanged, the binding and brazing being as originally finished.

Machinery Vibration.—Absence of machinery vibration is a point of considerable importance, especially in the case of a passenger steamer, and this advantage may be fairly laid claim to for turbine engines, the machinery vibration being practically *nil*, although the propellers often set up severe vibrations aft just over the stern.

R.M.S. "Viper" (see illustrations facing page 182).—The propelling machinery consists of three compound turbines, one high pressure and two low pressure; each of these turbines drives a separate shaft. The high-pressure turbine is on the centre shaft and the low-pressure turbines on the outer or wing shafts; with the latter are incorporated the reversing turbines which work in vacuum when the ship is going ahead; the reversing turbines can stop the vessel when going at full steam ahead in one and a half minutes from the time that the engineer receives the order from the captain.

The vessel has four double-ended boilers 20 ft. 6 in. long and 14 ft. in diameter, constructed for a working pressure of 165 lbs. per square inch. The air pumps and boiler feed-pumps are supplied by Messrs G. & J. Weir Limited. The bunkers have a capacity for 120 tons of coal.

NOTE.—On the trial runs over the measured mile the "Viper" attained a speed of fully 22 knots, with an air pressure of only $\frac{1}{2}$ inch water, while on the Cloch to Cumbrae and back continuous run the mean speed was fully 21 knots, with natural draught and easy steam.

Causes of Breakdown.—The principal causes of breakdown in turbine machinery may be classed as follows:—

1. **Stripping of Dummy Rings.**—This may be brought about by the (*a*) spindle bearings wearing down; or (*b*) the rotor out of position longitudinally; (*c*) insufficient clearance when heated up.

2. **Stripping of Blades.**—This may be brought about by—(*a*) Wear down of spindle bearings. (*b*) Insufficient clearance between rotor blades and case, or between casing blades and rotor when heated up. It should be noted that the expansion of the rotor

blades and rotor when heated up is much more than that of the rotor casing, thus reducing the blade clearance considerably, and increasing the risk of the rotor blades fouling the inside of the turbine casing. (*c*) *Water hammer.*—If a body of water is introduced into the turbine casing through priming in the steam pipes, the blades of the first and second expansion may be shattered by the resulting "water-hammer" action.

Up to the present there have been no cases recorded of "whipping" or "sagging" of the turbine rotor which has resulted in stripped blades, although this breakdown has been confidently predicted by many engineering experts.

The absolute importance of the regular oil supply to each of the rotor spindle main bearings cannot be over-estimated, as most of the cases which have actually occurred of dummy ring and blade stripping have been brought about by the temporary stoppage of the oil service, resulting, as may naturally be expected, in rapid heating up of the main bearings, melting of the white metal, and consequent wear down of the rotor, producing as a direct result the fouling of blades and stripping of some of the expansions. It is now the usual practice to insert a sight feed glass in the oil supply pipes, to enable the engineer on watch to see that the oil service is in working order, and the supply pipes all clear.

Corrosion in Rotor Drums. — One of the most serious troubles yet experienced in marine turbines is that of corrosion inside of the rotor drums. This is undoubtedly due to moisture and oxygen from air introduced into the turbine either from the water itself or more likely by leakage at the steam glands, which in the case of the L.P. turbines aft draw in the air, if not blowing out steam.

It should be observed that aft the L.P. turbines work in a high degree of vacuum; the after steam glands thus may draw in air, which, combining with the cast iron, produces by chemical action iron oxide on the inside of the cases. Again, the H.P. turbine runs in a vacuum when the two outside turbines only are running, so that under this condition air may also be drawn into the turbine casing, and corrosion take place.

The insides of the rotors are sometimes painted over with composition to check the corrosion referred to.

To prevent the admission of air it is advisable to see that the steam gland outer pocket cock is so regulated as to allow a constant puff outwards of weak steam when the turbines are running.

Wear of Dummy Rings.—In one or two cases it has been found that the dummy rings have been worn away to some considerable extent by the grinding action of chemical matter (iron oxide) produced by corrosion in the dummy cases. This may be assumed to have been brought about by galvanic action set up between the brass rings and cast-iron case when in contact in water, the chemical matter thus formed probably grinding away the rings.

The water which accumulates is produced by condensation in the dummy grooves, and constitutes the so-called "water seal" of the dummy. The corrosion referred to is a matter of much importance, affecting as it does the mechanical balance of the turbine, not to mention the resulting loss of economy.

Heating Up.—Previous to starting up a turbine the following precautions should be taken :—

1. Open all turbine casing drains, with "wet" air pumps working slow.
2. Admit steam (reduced) to outer pockets of steam glands until the gauge shows a pound or two pressure.
3. Test dummy clearances before heating up and after heating up. (This is important.)

24. Type—Channel Steamer.

H.P. Turbine.—On the cover being lifted, after about four months' service, signs of rusting were visible at the dummy "leak-off" port, also in dummy case. After dummy rings slightly worn, evidently due to contact, the "undercut" being much reduced and the corrosion decreasing in extent from the after to the forward rings.

L.P. Turbines.—Oxide of iron formation at "outer pocket" steam space, probably due to air (oxygen) finding its way into the turbines when working at low pressures. The inner pocket showed no signs of corrosion. The gland cases were more or less grooved showing that the brass rings were travelling round with the spindles. Four of these rings were broken in two places, and others were worn very thin (see sketch).

Signs of corrosion were evident at the reverse dummy fins on each side of each rotor fin and case fin. Some of these fins showed signs of actual contact.

The starboard L.P. ahead showed, on two sets of blades, severe oxidation; the dummies also gave similar evidences of oxidation. The S.L.P. dummy rings had evidently been in actual contact with the grooves, as these grooves were slightly recessed sideways, and the brass dummy rings themselves were broken off at intervals, and required to be renewed in the works. Although the finger plate indicated a clearance of about $\frac{20}{1000}$ in. when running, the faces must have been in actual contact, the reading being incorrect. This false indication may have been due to unequal expansion, when heated up, of the rotor, and the forward end of the casing, to which the finger plate is pinned down.

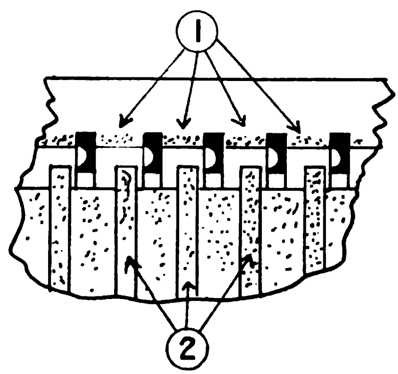
25. Type—Fast Cross-Channel Steamer.

L.P. Turbines.—After four months' service, corrosion (iron oxide) was evident inside the dummy cases, particularly at fillet of flange,

also on the rotor drum near the reverse fins. In the cylinder signs of corrosion were evident between the blade rows and near the leak-off port.

The starboard L.P. ahead dummy when tested showed a variation in clearance at each side of from $\frac{10}{1000}$ in. to $\frac{15}{1000}$ in. This was evidently caused by distortion of the casing due to unequal expansion by heat. The repair job was to grind up the dummies in place. The thrust rings and collars indicated no actual contact as the tool marks were still plainly visible. The gland rings were slightly worn, some showing a small ridge near the outer circumference, thus indicating that the rings had not been moving round with spindle, as is often the case, although not intended to be so. The rings showed signs of red oxide of iron, which, however, was quite easily removed with the finger.

H.P. Turbine.—The H.P. turbine showed signs of corrosion on the dummies, both casing and rotor. Oxide of iron deposits were



1. Pitting on Casing Dummy. 2. Pitting on Rotor Dummy.

found in fairly large quantities between the brass rings and at the bottom of the dummy piston grooves; at some positions the oxide quite filled up the space between the brass rings. The corrosion was most evident forward and diminished aft. In the opinion of the writer it may yet become necessary to construct both dummy case and piston of brass to reduce the corrosion referred to, which may reasonably be considered of a serious nature and requiring a drastic remedy.

In the foregoing case priming had been going on for some time and accounted for most of the corrosion described. It should also be mentioned that the "eccentric" dummy gauge and "poker" gauge both became stuck fast and unworkable, the point of the poker gauge also slightly corroded, which would, of course, upset the reading of the clearance.

(3.) Cover lifted after twelve months' service. No sign of wear is observable on thrust or main bearings, but both L.P. turbines show what is evidently the effects of steam and water friction, as at the

initial end of the rotor drums on the 1st expansion the surface is polished aft of each rotor blade row; this shows at the 1st, 2nd, and 3rd rows of the rotor 1st expansion. The burnishing referred to appears to have been caused by the water particles contained in the steam admitted to the L.P. turbines. The polished surfaces strongly resembled the effects of metallic contact, which at the positions noted was absolutely impossible.

26. Wear-down of Main Bearings.

(After four months' running.)

Type—Cross-Channel Steamer (22 Knots).

Bearing.	Amount Down.
H.P. forward	.003 in.
H.P. aft	.002 in.
P.L.P. forward	.002 in.
P.L.P. aft	.001 in.
S.L.P. forward	.002 in.
S.L.P. aft	.002 in.

The maximum wear-down was only $\frac{3}{1000}$ in. (.003 in.) for one season's running.

Finger-Plate Readings.—The finger-plate readings are occasionally so indexed that when the thrust surfaces are in close contact or "hard up" the finger plate's reading is less than the actual dummy clearance, which arrangement provides a margin of safety to work on. The following readings from a cross-channel steamer will perhaps make this clear:—

S.L.P. Dummy.—Half-rings, .300 in. forward and .270 in. aft taken out; half-rings, .310 in. forward and .260 in. aft put in, which gave readings of .018 in. on finger plate and .024 in. on dummy (by actual measurement on dummy rings). This gives a safety margin of (.024 – .018) = .006 in. or $\frac{6}{1000}$ in. Therefore, when even a $\frac{1}{1000}$ in. feeler could not be inserted between the finger plate and shaft groove, the dummies would still be clear by $\frac{6}{1000}$ in. or .006 in.

P.L.P. Dummy.—Hard up on thrust, .018 in. clear on finger plate and .026 clear on dummy.

H.P. Turbine.—Hard up on thrust, .009 clear on finger plate and .012 clear on dummy.

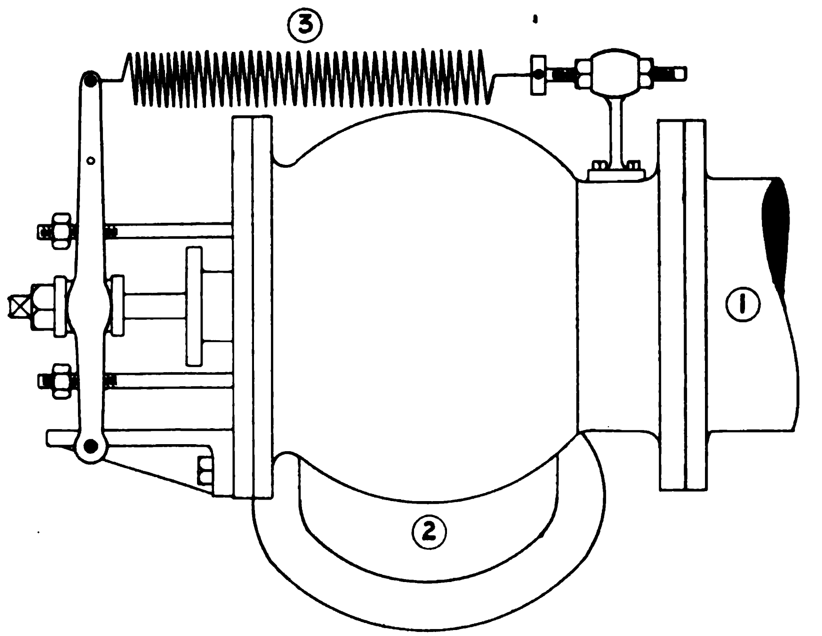
Propeller Vibration.—It has been found that the vibration aft, just over the propellers, is intensified—(1) When running in shallow water; (2) When rolling, on that side with the propeller nearest the surface of the water.

Steam Thrust and Propeller Thrust.—In the case of a fast channel steamer the L.P. dummy clearance varied about $\frac{8}{1000}$ in. or

.008 in. between an initial pressure of 10 lbs. and 16 lbs., this difference being caused by the propeller thrust exceeding the steam pressure thrust on the blades at or under 10 lbs. pressure and the steam thrust being in excess of the propeller thrust at and over 16 lbs. steam pressure.

Working of Turbines.

Going Ahead.—In going ahead full speed, steam is admitted by the main steam valve to the H.P. turbine, which it enters and



Spring-Loaded Non-Return Valve.

1. Exhaust from H.P. Turbine. 2. Admission to L.P. Turbine.
3. Pair of Springs.

This valve is fitted on the L.P. end of the H.P. exhaust pipes, and is intended to prevent the return of steam to the H.P. turbine when running with the L.P. turbines only.

passing through the rings of guide and shaft blades alternately and rotating the shaft, exhausts at the other (after) end to the two L.P. turbines, one on either side.

Entering the L.P. turbine casings, the steam passes through the various rings of blades, and finally exhausts at a very low pressure into the condensers. The independent air and circulating pumps maintain the vacuum, and Weir's or other patent feed-pumps deliver the feed-water into the boilers after it has passed through the feed-heater and feed-filter as commonly arranged. It should perhaps be

pointed out that the feed-heater fitted is often of the "surface" type —that is, one supplied with tubes through which the heating steam is blown, the water circulating through the chamber outside of the tubes. This arrangement prevents oily matter becoming mixed up with the feed-water when steam from the various auxiliary engines is used as heating steam for the heater.

Going Astern.—In going astern the main steam valve is closed and full-pressure steam is admitted by two valves to the astern turbines placed aft in the L.P. casings. When this is done, two large non-return valves, each fitted with a spring and placed between the H.P. and L.P. turbines, close automatically, and prevent the return of steam from the L.P. turbine back into the H.P. turbine.

The H.P. turbine is then in a vacuum, and the shaft and blades revolve idly as the propeller is set in motion by the water acting on the blades. Very little power is absorbed in this, as the vacuum offers but slight resistance to the rotation of the H.P. turbine blades.

Manœuvring.—As before stated, full-pressure steam can be admitted to the two L.P. turbines to drive ahead, the main steam being closed, or can be admitted to the two reverse turbines on the L.P. shaft to drive astern, the main steam valve still being closed, and the H.P. turbine revolving easily in a vacuum. It is also possible to give steam to one L.P. ahead turbine and the other L.P. reverse turbine at the same time, to allow of quick turning of the vessel or for manœuvring at piers, &c.

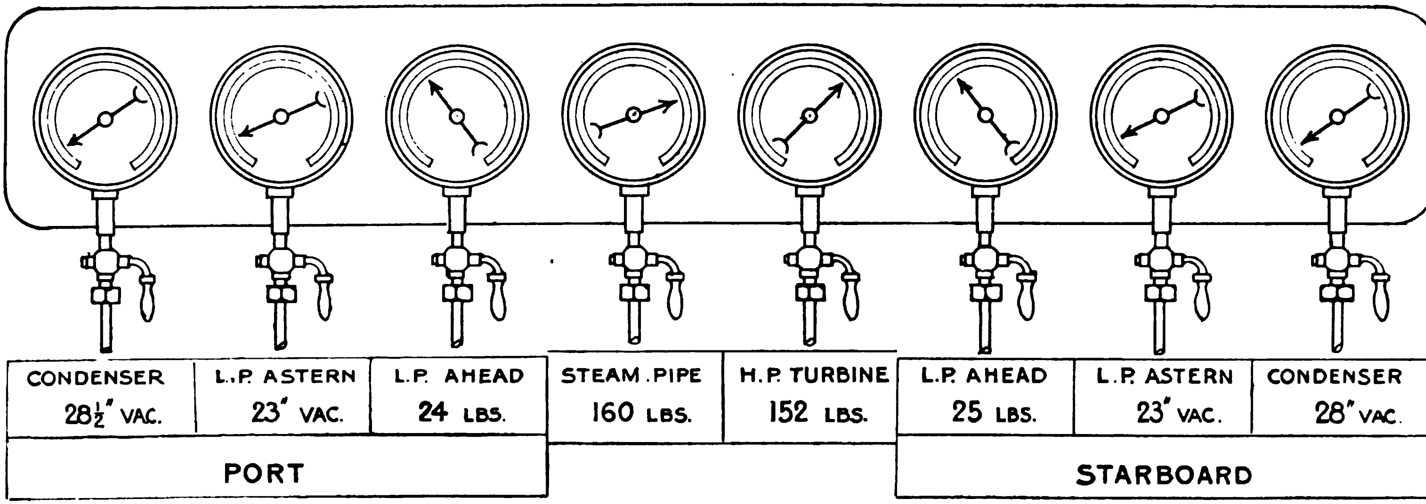
It will be obvious from the above that when going ahead with the two L.P. turbines, the L.P. reverse turbines are then running in a vacuum, and when going astern with the L.P. reverse turbines the L.P. ahead turbines are running in a vacuum.

Reversing.—Contrary to the first impressions and general expectations, the turbine steamers have proved that quick stopping and reversing can easily be accomplished if the reverse turbines are made of sufficient power. The "Dieppe" (cross-channel steamer) has supplied record figures for quick stopping and going astern, so that further doubt on this point is out of the question.

It should be stated that sudden reversing is very severe on the turbines, as excessive vibrations are set up by the change of rotation before the vessel comes to a stop.

Speed Regulation.—The engineer in charge regulates the speed as indexed by the telegraph, entirely by means of the revolution counters and the pressures shown on the various gauges and dials in connection with the turbine casings. Gauges are connected to the following points:—

1. Main steam pipe.
2. H.P. turbine.
3. Port L.P. turbine.
4. Starboard L.P. turbine.
5. Port L.P. astern turbine.
6. Starboard L.P. astern turbine.
7. Starboard condenser.
8. Port condenser.



Turbine Gauge Readings.

The above gives a fair idea as to the usual indications on the engine-room gauges of a large turbine steamer.

NOTE.—The gauges from No. 2 to No. 8 are of the "compound" type, and indicate either pressure or vacuum as required, according to the working of the various turbines.

Other gauges are fitted to show the oil pump pressure, cooling water pump pressure, &c. &c.

Number of Blades per Expansion.

In examining a list of blade numbers per row, or per expansion, it appears that, generally, the number of blades in each expansion of the casing is less than in each expansion of the rotor, although this does not hold good in all cases.

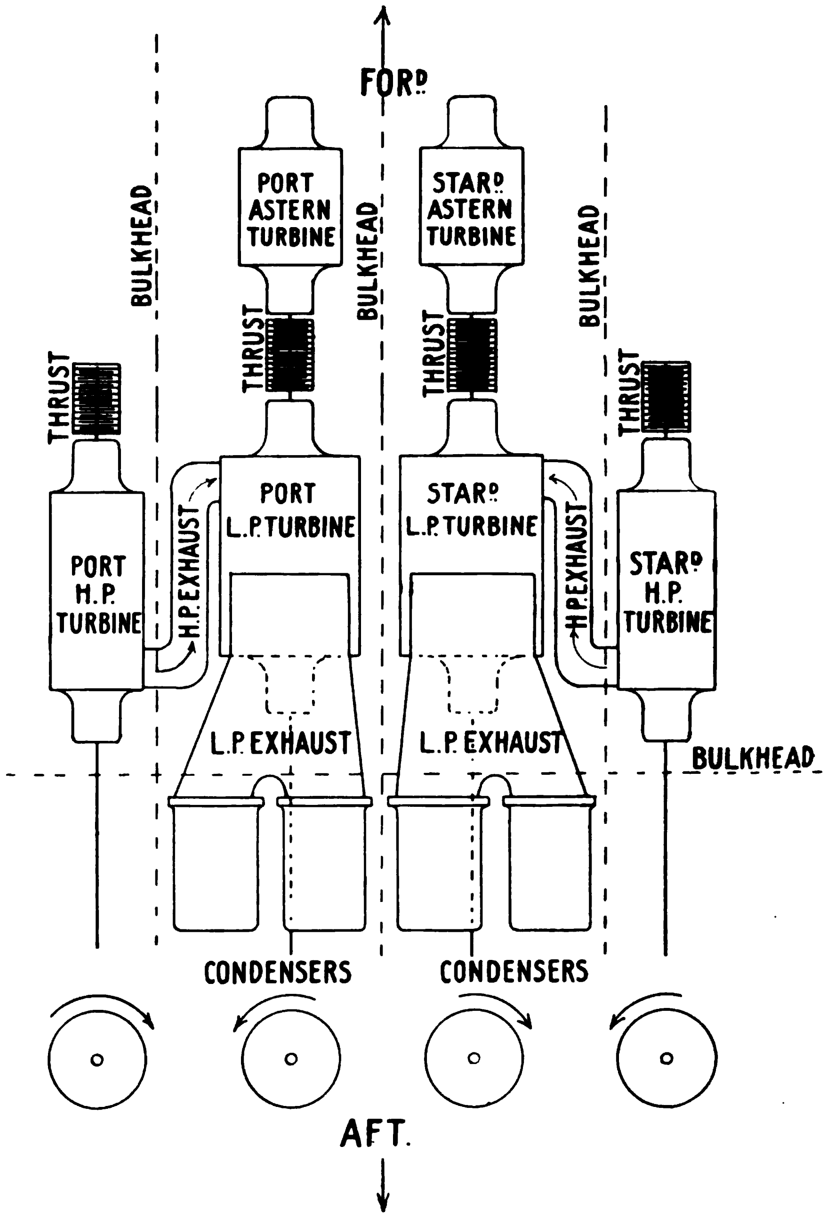
Case No. 1.—Type: Moderate Size Steamer.—The number of blades per row of the rotor in the first four expansions of the low-pressure turbines exceeds the number in the first four expansions of the cylinder, but from the 5th expansion on to the 8th expansion the number of blades per row in the cylinder exceeds the number in the rotor. In the H.P. and astern turbines the number of blades in the rotor exceeds the corresponding number in the cylinder throughout the four expansions of these turbines. Again, in the H.P. turbine the number of blades per row is the same for each expansion, but in the cylinder the number of blades per row *increases* from the 1st to the 4th expansions, the ratio of blade numbers being 1, 1.08, 1.16, and 1.2.

Referring to the astern turbines the number of blades per row is equal for the first six expansions of the rotor, but the 7th or last expansion has a smaller number. Again, in the astern cylinder the number of blades per row gradually increases from the 1st to the 6th expansion, the 7th expansion having a smaller number per row as in the case of the rotor.

Case No. 2.—Type: Small Steamer.—H.P. Turbine.—On referring to the H.P. turbine the number of blades per row in the rotor increases from the 1st expansion to the 3rd, and then drops away at the 4th, while in the cylinder or casing the number per row is constant for the 1st, 2nd, and 3rd expansions, and less for the 4th expansion. It is also noteworthy that the number of blades is less in the cylinder than in the rotor for the first three expansions but more for the 4th or last expansion.

L.P. Turbines.—In the rotor the number of blades per row increases from the 1st to the 3rd expansions. At the 4th expansion the number is much less, and from the 5th expansion to the 8th expansion the number gradually increases. In the cylinder the number of blades per row is constant for the first three expansions, then less in number and constant from the 4th expansion to the 7th expansion, the 8th or last expansion having a lesser number.

It is noteworthy that at all of the expansions except the 4th, the number per row in the cylinder is less than in the rotor, but at the 4th



Turbine Arrangement of "Lusitania" and "Mauretania."

The circles and arrows show the direction of shaft and propeller rotation, which is:—

Port H.P. and Starboard L.P. Right-hand Propellers.
Starboard H.P. and Port L.P. Left-hand Propellers.

expansion the number in the cylinder exceeds that of the rotor in the proportion of 566 to 690 blades. This is exceptional, as generally the number of blades in the cylinder is less than the number in the rotor throughout all the expansions.

Astern Turbines—Rotor.—The blade number per row decreases from the 1st to the 2nd expansion, then increases in the 3rd expansion, but again falls away in the 4th expansion.

Cylinder.—The number per row is equal in the 1st and 2nd expansions (but less than the number in the rotor), and in the 3rd and 4th expansions the number decreases.

The number in the first two expansions of the cylinder is less than in the corresponding rotor expansions, but in the 3rd expansion of the cylinder the number is *more* than in the rotor, while at the 4th expansion the number is *equal* in rotor and cylinder.

Case No. 3.—Type: Large Fast Steamer.—Rotor.—The 1st and 2nd L.P. rotor expansions have equal numbers of blades, the 3rd expansion less, the 4th expansion a few more than the 3rd, the 5th and 6th expansions equal numbers of blades but each less than the 4th expansion, the 7th expansion less blades than the 6th, and the last or 8th expansion less blades again than the 7th.

Cylinder.—In the L.P. cylinder the first two expansions have equal numbers of blades, but the number is less than those of the corresponding rotor expansions. The 3rd expansion has less blades than the 2nd, the 4th more than the 3rd, the 5th and 6th less than the 4th, and the 7th and 8th expansions less blades still, the numbers falling gradually.

The last four expansions have blades of equal height but of increasing angle and therefore exit area. The angle of the last expansion is somewhere between 65 and 80 degrees. It is also worth pointing out that the 4th expansion, in addition to being fitted with a greater number of blades than the 3rd expansion, has, together with the 5th and 6th expansions, the greatest number of rows of blades per expansion.

27. Trial Results.

Runs.	Boiler Pressure.	H.P. Receiver Pressure.	P.L.P. Turbine Initial Pressure.	S.L.P. Turbine Initial Pressure.	Condensed Vacuum.		Vacuum in P.L.P. Astern Turbine.	Vacuum in S.L.P. Astern Turbine.
					P.	S.		
1	130	85	7	4	28″	28″	17	22
2	155	140	30	25	25	28	17	22

28. Gland Steam Pressures.

	H.P. Forward End.	H.P. After End.	P.L.P. Forward End.	P.L.P. After End.	S.L.P. Forward End.	S.L.P. After End.
1	3	7	3	3	3	3
2	3	8	3	3	3	3

29. Consumption, Speed, and Power.

Boiler steam - - - - - - -	147 lbs.
H.P. steam - - - - - - -	120 „
Condenser vacuum - - - - - -	28 in.
Revolutions per minute - - - -	535
Steam used by turbines per hour - -	45,300
Steam used by auxiliaries per hour - -	6,000
Total evaporation per hour - - -	51,300
Speed - - - - - - - - -	15 knots
I.H.P. - - - - - - -	3,000
Steam used per I.H.P. hour (all purposes) -	$51{,}300 \div 3{,}000 = 17.1$ lbs.

30. General Data.

A.

Boiler steam - -	147 lbs.
H.P. turbine steam -	83 „
Condenser vacuum -	28 in.
I.H.P. (deduced) -	1,831
Speed - - - -	13.5 knots
Revolutions per min.	460
Coal per hour - -	4,500 lbs.
Evaporation per hour	35,500 „
Coal per I.H.P. hour (all purposes) -	2.25 „
Steam ditto - -	17.5 „
Evaporation per lb. coal - -	7.8

B.

Boiler steam - -	148 lbs.
H.P. turbine steam -	95 „
Condenser vacuum -	27.5 in.
Revolutions per minute P. -	435
Revolutions per minute C. -	486.6
Revolutions per minute S. -	415.6
Speed - - -	13.5 knots
Coal per hour - -	4,840 lbs.

NOTE.—With H.P. steam shut off and steam admitted to L.P. turbines only at 15 lbs. gauge pressure, the speed was 12.25 knots, and the revolutions, P 469, and S 439 per minute. The coal consumed was 5,670 lbs. per hour.

31.

	A.	B.	C.	D.
Boiler steam -	145 lbs.	146 lbs.	147 lbs.	145 lbs.
H.P. turbine -	100 „	125 „	115 „	100 „
L.P. turbines -	0 „	3 „	3.5 „	1 „
Condenser vac.	27.5 in.	27.25 in.	26.25 in.	27.25 in.
Revs. (mean) -	490 per min.	480 per min.	492 per min.	455 per min.
Speed - -	14 knots	13.25 knots	13.75 knots	12.5 knots
Sea temperature	74 degs.	84 degs.	90 degs.	79 degs.

	E.	F.
Boiler steam - - - - - -	155 lbs.	155 lbs.
H.P. turbine - - - - - -	150 „	149 „
L.P. turbines - - - - - -	8 „	8 „
Condenser vacuum - - - - -	27 in.	26.75 in.
Revolutions (mean) - - - - -	560 per min.	560 per min.
Speed - - - - - - - -	15.5 knots	15.75 knots
Sea temperature - - - - -	78 degs.	82 degs.
Condenser temperature - - -	103 „	102 „
Steam consumption in one hour -	61,200 lbs.	59,400 lbs.
Coal burnt in one hour at an evaporation of 7.8 lbs. - - - - -	7,846 „	...
Do. 8 lbs. - - - - -	...	7,425 lbs.

32. Test Runs showing effect of increase of Vacuum on Revolutions with constant Steam Consumption.

	H.P. Turbine Pressure.	L.P. Turbine Pressure.	Condenser Vacuum.	Sea Temperature.	Revolutions per Minute. H.P.	P.	S.	Mean.	Per cent. increase of Revolution Speed.
	Lbs.	Lbs.	In.	Degrees.					
A	88	0	25	81	455	411	359	408	1.96 °/ₒ for 1″ vac. (A–B)
B	88	½	26	81	454	424	369	416	
C	88	1	27	81	451	438	384	424	3.92 °/ₒ for 2″ vac.
D	145	4.5	25	78	560	484	473	506	
E	145	4.5	26	78	562	504	491	519	4.2 °/ₒ for $1\frac{3}{4}$″ vac.
F	145	4.5	26.5	78	562	509	499	523	
G	145	4.5	26.75	78	562	516	506	528	

NOTE.—In cases No. 29, 30, and 31 the form of the vessel was abnormal, and accounted chiefly for the low power efficiency and low speed shown.

33.

Boiler steam - - - - - - -	148 lbs.
H.P. turbine - - - - - - -	145 „
L.P. turbines - - - - - -	6 „
Condenser vacuum - - - - - -	P. 26 in., S. 26¼ in.
Sea temperature - - - - - -	86 degs.
Condenser temperature - - - - -	P. 112 degs., S. 108 degs.
Revolutions (mean) - - - - - -	537 per minute.
Speed - - - - - - - -	15.5 knots.
Slip (mean) - - - - - - -	23 per cent.
Finger-plate reading of dummy clearance -	P. .013 in., C. .015 in., S. .025 in.

34. Finger-plate Readings of Dummy Clearance, showing effect of damaged Port Propeller.

	P.	C.	S.	
	In.	In.	In.	
A. Finger-plate readings (normal) - -	.018	.017	.019	... ...
B. Do., with P. propeller twisted - - -	.025	.014	.018	Revolutions fell off 35 below normal.
C. Do., with bye-pass open - - -	.026	.014	.018	... ...
D. Do., with new port propeller (bye-pass shut) - - -	.017	.014	.019	... ...

NOTE.—It will be obvious that if the propeller falls off in propulsive efficiency, and acts to some extent as a drag on the hull, then the dummy clearance will be increased, as the rotor position is now further aft than formerly.

Report of Turbine Overhaul.—"Starboard L.P. turbine opened up; rotor lifted, and six rows of 1st expansion found badly choked with hard, dry grit; first row of 2nd and 3rd expansions badly clogged with greasy matter from blade roots up to half height, but clear above this; all blades more or less coated with dirty deposit. The cylinder blades were found to be in the same condition as the rotor blades, but the greasy deposit was heaviest near blade tips, evidently indicating the tendency of the steam to flow outwards from the centre."

NOTE.—In the foregoing case priming had been going on for some time with the results stated.

Dummy.—The starboard L.P. astern dummy radial fins were found burred on the edges more aft than forward; the burrs were removed and the "fin" edges recut. The blade tip clearances (taken cold) were as follows:—

35. Blade Clearances (taken cold).

Expansions.	1.	2.	3.	4.	5.	6.
Rotor - -	.040 in.	.046 in.	.053 in.	.060 in.	.065 in.	.083 in.
Cylinder -	.031 "	.037 "	.047 "	.050 "	.055 "	.060 "

36. Data for Reduced Speeds.

	A.			B.		
Boiler steam - - - -	185 lbs.			185 lbs.		
H.P. turbine initial pressure -	50 "			18 "		
H.P. turbine terminal pressure	$2\frac{1}{2}''$ vac.			17.5″ vac.		
P.L.P. initial pressure - -	3″ "			18″ "		
S.L.P. initial pressure - -	3″ "			20.5″ "		
L.P. astern turbines - -	26″ "			S. 26.5″ vac.		P. 27″ vac.
Condensers - - - -	$28\frac{1}{2}''$ "			29″ vac.		
	H.P.	S.L.P.	P.L.P.	H.P.	S.L.P.	P.L.P.
Revolutions per minute - -	310	325	320	225	235	230
Speed - - -	15.5 knots.			11 knots.		
	H.P.	S.L.P.	P.L.P.	H.P.	S.L.P.	P.L.P.
Dummy clearance - - -	.012″	.026″	.025″	.014″	.030″	.032″

Note.—It will be observed that the dummy clearances increase with the decrease of speed.

Trial Results.

37. Type—Fast Channel Steamer.

	A.			B.		
Boiler steam	175 lbs.			175 lbs.		
H.P. turbine initial pressure	156 „			154 „		
H.P. turbine terminal pressure	24 „			24 „		
P.L.P. turbine initial pressure	22 „			$22\frac{1}{4}$ „		
S.L.P. turbine initial pressure	23 „			$22\frac{3}{4}$ „		
Condenser vacuum	P. 28.1 in.		S. 28.1 in.	P. 28.2 in.		S. 28.2 in.
	P.L.P.	H.P.	S.L.P.	P.L.P.	H.P.	S.L.P.
Revolutions per minute	493	498	505	491	496	505
Slip	21 per cent. (mean).					
	H.P.			Wing.		
Diameter of propellers (3 blades)	6 ft. 9 in.			6 ft. 9 in.		
Pitch of propellers (H.P. and S.L.P. right hand; P.L.P. left hand)	5 ft. 10 in.			6 ft. 1 in.		
Expanded surface	20.5 sq. ft.			20.5 sq. ft.		
Projected surface	17.8 „			17.6 „		

38. Dummy Clearance—Case A.—The following data (taken specially for the writer) refers to dummy clearances, taken (1st) when cold (2nd), when "heating up," and (3rd) when actually running.

DUMMY CLEARANCE.

	H.P.	P.L.P.	S.L.P.
	Inch.	Inch.	Inch.
Taken cold	.031 or $\frac{31}{1000}$	.030 or $\frac{30}{1000}$	.027 or $\frac{27}{1000}$
Taken when "heating up"	.020 or $\frac{20}{1000}$	.040 or $\frac{40}{1000}$	.043 or $\frac{43}{1000}$
Taken when running	.012 or $\frac{12}{1000}$	.027 or $\frac{27}{1000}$	.029 or $\frac{29}{1000}$

NOTE.—It will be observed that in the above case the H.P. dummy clearance becomes less when the turbines are being heated up, and less again when

the full expansion due to heat acts on the metal as when actually running, whereas the L.P. turbines show an increased clearance when heating up which is again decreased when running. This is due to the heavier material of the larger L.P turbines expanding more than the H.P., and acting to reduce the clearance by expanding the rotor in excess of the cylinder.

Case B.

	H.P.	P.L.P.	S.L.P.
	Inch.	Inch.	Inch.
Cold - - -	.026	.018	.025
Heating up -	.030	.031	.034
Running - -	.021	.021	.025

In the above case the dummy clearance increases in each turbine when heating up but decreases again when running.

Case C.—The variations in dummy clearance due to different conditions of working and running are shown in the following cases from actual practice:—

	Cold.	Heating up.	When Running.	Working with L.P. Turbines only.	Going Astern.
	In.	In.	In.	In.	In.
H.P. Dummy clearance -	$\frac{50}{1000}$	$\frac{30}{1000}$	$\frac{20}{1000}$	$\frac{15}{1000}$	...
L.P. turbine dummy clearance	...	...	...	...	$\frac{60}{1000}$

Case D.

	H.P.	P.L.P.	S.L.P.
	In.	In.	In
Cold - - - -	$\frac{28}{1000}$	$\frac{45}{1000}$	$\frac{43}{1000}$
Heating up - -	$\frac{17}{1000}$	$\frac{35}{1000}$	$\frac{32}{1000}$
Running at full speed -	$\frac{22}{1000}$	$\frac{40}{1000}$	$\frac{38}{1000}$
Astern - - -	...	$\frac{46}{1000}$	$\frac{45}{1000}$

In these turbines the clearances are slightly less than the above when running at reduced speed.

39. Type—Fast Channel Steamer.

Speed, 22 knots.
Equivalent I.H.P., 8,500 (approximate).

Turbine Data.

H.P. Turbine. (Drum, 2 ft. 6 in. Diam.)

Expansion.	Number of Blade Rows	Blade Heights.
1	13	$1\frac{1}{2}$ in.
2	13	$2\frac{1}{4}$ "
3	14	3 "
4	14	$4\frac{1}{2}$ "

L.P. Turbines. (Drum, 3 ft. 9 in. Diam.)

Expansion.	Number of Blade Rows.	Blade Heights.
1	7	$1\frac{1}{8}$ in.
2	7	$2\frac{1}{4}$ "
3	7	3 "
4	7	$4\frac{1}{2}$ "
5	7	6 "
6	7	8 "
7	7	8 "
8	7	8 "

NOTE.—6th, 7th, and 8th expansions "semi-wing blades."

Expansion.	Number of Blade Rows.	Blade Heights.
1	10	$\frac{11}{16}$ in.
2	10	$2\frac{1}{2}$ "
3	10	$2\frac{7}{8}$ "
4	10	$2\frac{7}{8}$ "
5	10	$2\frac{7}{8}$ "

NOTE.—4th and 5th expansions "wing blades."

H.P. dummy consists of twenty rings aft of "leak-off" space, and fourteen rings forward of "leak-off" space.

L.P. dummies consist of fourteen rings aft of "leak-off" space, and six rings forward of "leak-off" space.

H.P. glands (old style) consist of thirteen rings aft, five rings in centre, and five rings forward.

Thrusts consist of twenty-three $\frac{5}{16}$-in. brass rings in block, and twenty-three $\frac{1}{4}$-in. collars on shaft, with $\frac{3}{8}$-in. recesses between.

Shafting, 7 in. diameter.

Trial Results.

Boiler pressure	150 lbs.
H.P. turbine	143 ,,
P.L.P.	19½ ,,
S.L.P.	19½ ,,
Condenser vacuum	P. 27 in., S. 27 in.
Revolutions (mean) per minute	666
Speed	22 knots.
Oil pressure (forced lubrication)	9 lbs.
Cooling water pressure (bearings)	8 ,,
Air-pump strokes per minute	P. 38, S. 41
Air pressure at fan	¼ in. water.
Heating surface (total)	16,400 sq. ft.
Grate surface	506 ,,
Cooling surface	6,448 ,,
Consumption per square foot of grate per hour	29 lbs.

40. Reduced Speed Data.

Initial pressures at H.P. expansions (engine steam, 173 lbs.):—

No. of Expansion.	Gauge Pressure.
1	34 lbs.
2	17 ,,
3	7 ,,
4	0 ,,
Terminal. 4	0 ,,

NOTE.—The H.P. gauges did not indicate below atmospheric pressure, hence the absence of figures for the 4th expansion initial and final pressures.

Initial pressures at L.P. expansions :—

1	9 in. vacuum.
2	15 ,, ,,
3	20.5 ,, ,,
4	23.5 ,, ,,
5	25.5 ,, ,,
6	26 ,, ,,
7	26.5 ,, ,,
8	27 ,, ,,
Terminal (condenser) 8	27.5 ,, ,,

41. Owing to reduced speed of starboard L.P. turbine due to the choking up of a number of blades, as a remedy the *port* non-return valve connecting the H.P. and port L.P. turbine was partly closed down, and this had the effect of raising the initial pressure in the starboard L.P. and increasing the revolutions of that turbine as the following data indicate. This adjustment somewhat resembles the "linking up" of a reciprocating engine valve gear.

A (before partly shutting port L.P. valve).

H.P. turbine	143 lbs.
L.P. turbines	8 "
Condenser vacuum	P. 26¾ in., S. 27 in.
Sea temperature	79 degs.
Revolutions	P. 564, C. 563, S. 499.
Speed	16 knots.
Dummy clearance by finger-plates	P. .023 in., C. .0125 in., S. .016 in.

B (after partly shutting port L.P. valve).

H.P. turbine	150 lbs.
S.L.P. turbine	12.5 "
P.L.P. turbine	4 "
Condenser vacuum (mean)	27 in.
Sea temperature	79 degs.
Revolutions per minute	S. 508, C. 536, P. 518.

NOTE.—H.P. exhaust to port L.P. turbine partly closed.

42. Initial Pressures at "Expansions."—The following initial and terminal pressures of the H.P. turbine "expansions" should prove of interest, the turbines to which the pressures refer being of large power :—

H.P. TURBINE (NO. 1).

Boiler steam	210 lbs. gauge.
1st expansion	120 "
2nd "	70 "
3rd "	50 "
4th "	35 "
5th "	12 "
6th "	3 "
6th (Terminal pressure)	1 "

H.P. TURBINE (NO. 2).

Boiler steam	220 lbs. gauge.
1st expansion	125 "
2nd "	80 "
3rd "	53 "
4th "	33 "
5th "	16 "
6th "	6 "
6th (Terminal pressure)	2 "

For the foregoing the coal consumption varied from 2 lbs. per I.H.P. per hour at $\frac{1}{5}$ power to 1.3 lbs. per I.H.P. per hour at full power and speed, constituting a very fine performance indeed. The oil pressure was 8 lbs. per square inch at the bearings, and the oil temperature at discharge from bearings 133 degs., and when going through oil filter 50 degs.

43. Type—Pleasure Yacht.

Horse-power	5,200
Trial speed	17.75 knots.
Revolutions per minute	500 (designed).
Boiler pressure	150 lbs.
Propeller pitch	5 ft.
Propeller diameter	6 ft.

44. Turbine Data.

H.P. rotor, 42 in. diameter ; L.P. rotor, 60 in. diameter ;
Reverse rotor, 42 in. diameter.

Expansions.	Blade Heights.		
	H.P.	L.P.	Reverse.
1	$\frac{9}{16}$ in.	$\frac{9}{16}$ in.	$\frac{3}{8}$ in.
2	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$\frac{3}{4}$ "
3	$1\frac{5}{8}$ "	$1\frac{5}{8}$ "	$1\frac{1}{2}$ "
4	$2\frac{1}{4}$ "	$2\frac{1}{4}$ "	$1\frac{1}{2}$ " *
5	...	$3\frac{1}{4}$ "	...
6	...	$4\frac{1}{2}$ "	...
7	...	$4\frac{1}{2}$ " *	...
8	...	$4\frac{1}{2}$ " *	...

* "Semi-wing" and "wing" blades.

Each H.P. expansion consists of 16 blade rows (64 in all).
" L.P. " " " 8 " " (" ").

45. H.M.S. "Dreadnought."

The following data are taken from the pages of the *American Journal of Naval Engineers* :—

Boilers, 18 Babcock & Wilcox type, fitted for burning oil fuel in conjunction with coal. Each boiler consists of 20 "elements."
Total grate surface, 1,560 sq. ft.
Total heating surface, 55,400 sq. ft.
Ratio of H.S. to G.S., 35.5 to 1.
Height of funnel above fire bars, 85 ft.

Designed consumption at full power, 34,500 lbs. of coal per hour.
Cooling surface of main condensers, 26,000 sq. ft.
Cooling surface of auxiliary condensers, 6,000 sq. ft.
No. of shafts, 4
No. of propellers, 4 (one per shaft).
On wing shafts, 1 H.P. ahead turbine and 1 H.P. astern turbine.
On centre shafts, 1 cruising turbine, 1 L.P. ahead turbine, and 1 L.P. astern turbine.

NOTE.—The wing propellers are placed forward of the centre propellers, as in the "Lusitania" and "Mauretania."

Turbines.

	Rotor Drums.	
	Diameter.	Length.
Cruising - -	5 ft. 8 in.	8 ft. $6\frac{3}{8}$ in.
H.P. ahead - -	5 ft. 8 in.	8 ft. $7\frac{1}{8}$ in.
H.P. astern - -	5 ft. 8 in.	3 ft. $1\frac{1}{8}$ in.
L.P. ahead - -	7 ft. 8 in.	6 ft. 6 in.

Blading.

	H.P. Rotors.	L. P. Rotors.	
No. of blade rows - -	72	36	
No. of expansions - -	6	6	
Blade heights—			
1st expansion - -	$\frac{7}{8}$ in.	$3\frac{7}{8}$ in.	
2nd " - -	$1\frac{1}{4}$ "	$5\frac{1}{2}$ "	
3rd " - -	$1\frac{3}{4}$ "	$7\frac{3}{4}$ "	
4th " - -	$2\frac{1}{2}$ "	11 "	Wing-blades.
5th " - -	$3\frac{1}{2}$ "	11 "	
6th " - -	5 "	11 "	

NOTE.—It should be noted that although the volume of steam entering the L.P. turbines is fully more than that of the steam exhausting from the H.P. turbines, the blade heights drop from 5 in. in the last H.P. expansion to $3\frac{7}{8}$ in. in the first L.P. expansion, and this is accounted for by the larger diameter, and therefore steam area annulus, of the L.P. turbines, which allow of much smaller blade heights for similar area of steam flow. This difference is much more accentuated in the case of the standard type three turbine arrangement, as two L.P. turbines being fitted only half the volume of steam exhausted from the H.P. turbines is admitted to each L.P.

Trial Results.

THIRTY HOURS' TRIAL AT 4,600 HORSE-POWER, 3RD AND 4TH OCTOBER 1906.

Steam pressure in boilers, lbs.	220.0		220.0		220.0		220.0	
Air pressure in stokeholds, inch	0.58		0.58		0.60		0.56	
Duration of trial, hours	Ten		Fifteen		Five		Means and totals	
	Starboard.	Port.	Starboard.	Port.	Starboard.	Port.	Starboard.	Port.
Steam pressure at engines, lbs.	210.0	210.0	210.0	210.0	210.0	210.0	210.0	210.0
Initial steam pressure in H.P. turbines, lbs.	34.5	30.2	25.5	22.8	31.6	28.0	29.5	26.1
„ vacuum in L.P. turbines, in.	13.3	16.1	9.6	12.6	11.4	15.4	11.1	14.2
„ steam pressure in cruising turbines	92.7	91.6	73.8	74.46	87.3	86.4	82.3	82.1
Vacuum in condensers, in.	27.85	27.75	27.73	27.48	27.3	27.48	27.7	27.57
Revolutions per min., inner shafts, number	197.94	194.76	215.82	207.89	205.82	196.95	208.2	201.69
„ „ wing shafts, „	191.12	186.27	178.12	175.58	185.74	183.4	183.72	180.45
Horse-power, inner shafts	1,516.6	1,469.2	2,000.0	1,781.0	1,698.8	1,489.0	1,788.6	1,630.3
„ wing shafts	999.77	971.02	650.0	656.0	854.1	861.8	800.41	794.03
„ total	2,516.37	2,440.22	2,650.0	2,437.0	2,552.9	2,350.8	2,589.01	2,424.33
„ grand total	4,956.59		5,087.0		4,903.7		5,013.34	
Steam condensed per horse-power—								
Per hour from main engines, lbs.	20.8 } 27.2		25.1		27.3		25.9	
„ „ auxiliary engines, lbs.	6.4							
Waste of feed water, tons	14.0		25.7		8.7 (gain)		31.0	
Water evaporated per lb. of coal, lbs.	10.3		10.28		9.98		10.2	
Coal per horse-power per hour, lbs.	2.6		2.42		2.74		2.6	
Exhaust steam from auxiliaries led into	Auxiliary condenser		On turbines		...		Various	
Speed of ship, by log, knots	13.0		13.0		13.0		13.0	

PROGRESSIVE TRIALS AT VARIOUS POWERS, 4TH OCTOBER 1906.

Steam pressure in boilers, lbs.	220		215		227		218		223		230		235		242	
Air pressure in stokeholds, in.	...		...		0.13		0.30		0.80		0.80		0.50		0.75	
Duration of trial, hours	Two		One		Two		One		One		Two		One		Two	
	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.
Steam pressure at engines, lbs.	208	208	201	201	218	218	208	208	215	215	217	217	224	224	227	227
Initial steam pressure—																
High-pressure turbines, lbs.	4.0	1.0	10.0	13.5	16.0	15.5	14.0	13.5	74.0	69.0	74.7	68.2	87.0	82.0	87.0	82.0
Low-pressure turbines, lbs.	Vac. 17 in.	Vac. 22 in.	Vac. 16 in.	Vac. 20 in.	Vac. 14 in.	Vac. 18.5	Vac. 12 in.	Vac. 15.7	Vac. 5 in.	Vac. 7 in.	Vac. 5 in.	Vac. 4 in.	0	Vac. 3 in.	1.25	0.39
Cruising turbines, lbs.	25	25	25	25	55	55	55	55	179	178	179	179	180	178	180	179
Vacuum in condensers, in.	28.0	27.8	28.0	27.0	28.0	27.8	27.1	27.4	275	27.6	26.7	27.8	26.5	27.5	26.2	27.5
Revolutions per minute—																
Inner shafts, number	117.0	111.4	128.8	122.1	158.4	152.5	181.5	176.3	263.7	261.1	289.4	279.2	277.4	275.8	306.1	301.7
Wing shafts, number	121.7	115.6	140.8	138.6	155.4	149.2	154.8	149.7	249.6	245.0	245.9	242.2	266.3	273.3	264.0	262.5
Horse-power, inner shafts	415	327	485	394	842	844	1 293	1,197	3,542	3,465	4,858	4,388	3,901	4,383	5,631	5,505
,, wing shafts	279	282	436	433	567	518	485	448	2,213	2,081	1,957	1,889	2,697	2,767	2,327	2,412
,, total	694	609	921	827	1,409	1,362	1,778	1,645	5,755	5,546	6,815	6,277	6,598	7,150	7,958	7,917
,, grand total	1,303		1,748		2,771		3,423		11,301		13,092		13,748		15,875	
Steam condensed per H.P. per hour—																
From main engines, lbs.	34.9	58.5	37.4	41.6	38.2	49.7	32.3		16.01	19.9	17.7	18.16	15.3	18.9	16.6	
From auxiliary engines, lbs.	23.6		4.2		11.5				3.9		0.46		3.6			
Exhaust steam from auxiliary led into	Aux. cond.		On turbines.		Aux. cond.		On turbines.		Aux. turbines.		On turbines.		Aux. cond.		On turbines.	
Speed of ship, by log, knots	...		...		...		...		...		...		...		...	

Thirty Hours' Trial at 16,250 Horse-Power, 6th and 7th October 1906.

The first ten hours without closed exhaust; the next six hours with closed exhaust on turbines and auxiliary condenser; the last fourteen hours with closed exhaust on turbines and evaporators.

Mean Results.

	Stbd.	Port.
Steam pressure in boilers	232 lbs.	
Air pressure in stokeholds	0.9 in.	
Duration of trial	30 hours.	
Initial steam pressures in—	Stbd.	Port.
High-pressure turbines	110.4 lbs.	109.3 lbs.
Low-pressure turbines	0.6 „	3.75 „
Cruising turbines	...	...
Vacuum in condensers	27.5 in.	27.9 in.
Revolutions per minute, inner shafts	298.34	294.4
„ „ wing shafts	286.22	286.08
Horse-power inner shafts	5,092	4,988
„ wing shafts	3,350	3,500
„ total	8,442	8,488
„ grand total	16,930	
Steam condensed per horse-power	17.01 lbs.	
Waste of feed water	54.03 tons.	
Water evaporated per pound of coal	10.01 lbs.	
Speed of ship	19.3 knots.	

Measured Knot Trial, 9th October 1906.

Run.	Revolutions per Minute. Starboard. Inr.	Starboard. Wing.	Port. Inr.	Port. Wing.	Time along Knot.	Speed.	Horse-power. Stbd. all.	Port all.	All of all.
					Min. Sec.	Knots.			
1	354.9	324.8	342.0	320.8	3 10	21.78	13,774	14,125	27,899
2	353.2	333.3	351.4	328.2	3 13	21.45	13,663	13,670	27,333
3	348.5	329.5	347.3	325.8	3 10	21.78	13,140	13,388	26,728
4	340.8	328.1	340.8	327.4	3 13.5	21.39	12,842	13,270	26,112
			Simpson's mean			21.6		Mean	27,018

EIGHT HOURS' TRIAL AT 23,000 HORSE-POWER, ON 9TH OCTOBER 1906.

MEAN RESULTS.

	Stbd.	Port.
Steam pressure in boilers	241 lbs.	
Air pressure	1.2 in.	
Initial steam pressures—	Stbd.	Port.
In cruising turbines	...	...
In high-pressure turbines	157.5 lbs.	154.5 lbs.
In low-pressure turbines, pounds	7.7 „	12.0 „
Vacuum	27.0 in.	27.4 in.
Revolutions per minute, inner shafts	337.2	333.3
„ „ wing shafts	322.2	321.7
Horse-power, inner shafts	7,430	7,447
„ wing shafts	4,795	5,040
Total	12,225	12,487
Aggregate horse-power	24,712	
Coal per indicated horse-power per hour	1.51 lbs.	
Water evaporated per pound of coal	10.03 „	
Total loss of feed water	25.1 tons.	
Water per horse-power—		
Main engines per hour	14.41 } 15.56 lbs.	
Auxiliary engines per hour	1.15 }	
Speed of ship	21.25 knots.	
Slip at $19\frac{1}{2}$ knots	21 per cent.	
„ full speed	26 „	

46. Report on Working of Turbine Drains and Steam Glands.—"The drain from H.P. turbine to L.P. port turbine is opened when the H.P. turbine is stopped, and shut (except for one turn) when the H.P. turbine is running. In port the H.P. drain is opened to bilges if air pumps are stopped.

"The drain on forward end of H.P. turbine has two connections, one to the after end of the same turbine and one direct to the bilges. The one on after end is opened when heating up and shut when all water is out of the pipes, &c.

"When heating up, steam is admitted to H.P. glands, but when running at full power this acts as a 'leak-off' to L.P. 3rd expansion. At low speeds and powers it is required to shut the 'leak-off' connection and admit steam to the gland to prevent the formation of a vacuum and danger of air admission to the turbine. The pressure on H.P. glands varied from 1 to 2 lbs., and on L.P. glands from 2 to 4 lbs."

NOTE.—In this steamer the dummy clearance cold was *less* than when heated up, whereas the reverse is generally the case.

Turbine Efficiency.—The over-all efficiency of a turbine may be determined as shown in the following case taken from actual practice :—

Data.

H.P. turbine, initial pressure - - - - -	173 lbs. (gauge).
L.P. turbines, terminal pressure - - -	26 in. vacuum.
Shaft horse-power - - - - - - -	17,560.
Coal per hour - - - - - -	12.5 tons.
Speed - - - - - - - - -	$20\frac{1}{2}$ knots.
Evaporation (assumed) - - - - -	8.6.

Then,

Initial $\left\{\begin{array}{l} 173 + 15 = 188 \text{ lbs. absolute} = 376.4 \text{ temperature Fahr.} \\ 376.4 + 461 = 837.4 \text{ absolute temperature.} \\ 188 \text{ lbs.} = 847 \text{ B.T.U. latent heat.} \\ \text{Dryness} = 1 \text{ (assumed).} \end{array}\right.$

Terminal $\left\{\begin{array}{l} 2 \text{ lbs. absolute} = 126.3 \text{ deg. temperature Fahr.} \\ 126.3 + 461 = 587.3 \text{ absolute temperature.} \\ 2 \text{ lbs.} = 1026 \text{ B.T.U. latent heat.} \\ \text{Dryness} = .778 \text{ (adiabatic expansion assumed).} \end{array}\right.$

$$\begin{aligned} \text{Heat drop} &= 847 \times 1 - 1026 \times .778 + 837 - 587.3 \\ &= 847 - 798.228 + 837.4 - 587.3 \\ &= 1684.4 - 1385.52 \\ &= 298.88 \text{ (say 299) B.T.U.} \end{aligned}$$

$$\text{Steam flow per minute} = \frac{12.5 \times 2240 \times 8.6}{60} = 4013 \text{ lbs.}$$

$$\text{Theoretical horse-power} = \frac{299 \times 778 \times 4013}{33000} = 28288 \text{ horse-power.}$$

Actual horse-power = 17560.

Therefore, $17560 \div 28288 = .62$ efficiency.

NOTE.—Steam consumption per H.P. hour $= \dfrac{33000 \times 60}{299 \times 778 \times .62} = 13.72$ lbs.

EXAMPLE.—Determine the theoretical heat drop at the 2nd H.P. expansion, the initial pressure being 112 lbs. (gauge) and the terminal pressure (or 3rd expansion initial pressure) 68 lbs. (gauge). Assume dryness factor of .97 and .95.

Then,

Initial $\left\{\begin{array}{l} 112 + 15 = 127 \text{ lbs. absolute} = 345.4 \text{ temperature and } 870.7 \text{ latent heat} \\ 345.4 + 461 = 806.4 \text{ absolute temperature. Dryness .97 (assumed).} \end{array}\right.$

Terminal $\left\{\begin{array}{l} 68 + 15 = 83 \text{ lbs. absolute} = 314.5 \text{ temperature and } 892.5 \text{ latent heat.} \\ 314.5 + 461 = 775.5 \text{ absolute temperature. Dryness .95 (assumed).} \end{array}\right.$

$$\begin{aligned} \text{Then, } & 870.7 \times .97 - 892.5 \times .95 + 806.4 - 775.5 \\ &= 844.579 - 847.87 + 806.4 - 775.5 \\ &= 1650.97 - 1623.37 \\ &= 27.60 \text{ B.T.U.} \end{aligned}$$

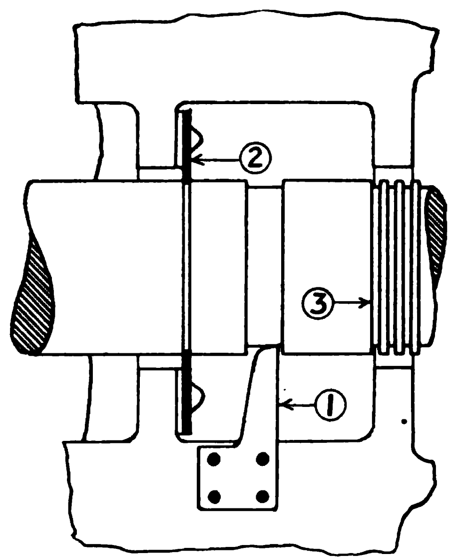
EXAMPLE.—Calculate the heat drop at the last L.P. expansion, the initial pressure being 22 in. vacuum (4 lbs. absolute) and the terminal pressure 26 in. vacuum (2 lbs. absolute). Dryness factors .8 and .79.

Initial $\left\{\begin{array}{l} 4 \text{ lbs. absolute} = 153.1 \text{ temperature Fahr. and } 1006.8 \text{ latent heat.} \\ 153.1 + 461 = 614.1 \text{ absolute temperature. Dryness .8.} \end{array}\right.$

Terminal $\left\{ \begin{array}{l} \text{2 lbs. absolute} = 126.3 \text{ temperature Fahr. and } 1025.8 \text{ latent heat.} \\ 126.3 + 461 = 587.3 \text{ absolute temperature. Dryness } .79. \end{array} \right.$

$$\begin{aligned} & 1006.8 \times .8 - 1025.8 \times .79 + 614.1 - 587.3 \\ & = 805.44 - 810.38 + 614.1 - 587.3 \\ & = 805.44 + 614.1 - 810.38 - 587.3 \\ & = 1419.54 - 1396.78 \\ & = 21.86 \text{ B.T.U.} \end{aligned}$$

Dummy Clearance Gauge.—A small flat plate, called a "finger plate," is pinned down to part of the lower casing frame at the forward end, and the edge of the plate projects against a collar on the turbine shaft, so that by means of a feeler or wedge the clearance between



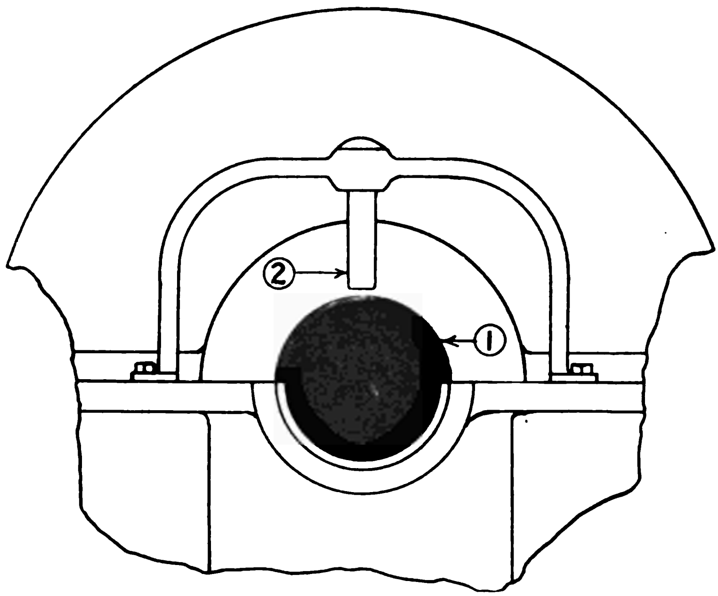
"**Finger-plate**" **Gauge.**

1. Finger Plate. 2. Oil Deflector. 3. Steam Gland Rings.

the two can always be known. This clearance is noted when the turbine is completed, and a record kept, so that any change of clearance due to wear fore or aft can at once be noted by future testing. Longitudinal change of position may occur through wear of the thrust block rings, and to take up this (as mentioned elsewhere) a screw is fitted on the top half of the thrust block cover with an adjusting nut, and half brass rings fitted for adjustment of the lower or ahead portion of the thrust. A set of these rings of varying thickness is supplied as spare gear, and can be inserted as required so as to maintain the same dummy clearance.

"**Bridge**" **Gauges.**—An appliance called a "bridge gauge" is used at each end of the rotor to test for possible wear down, which, as

will readily be understood, is a matter of paramount importance. The gauges merely consist of a piece of bent round bar iron with a centre boss through which is screwed and riveted a hardened steel pin with a flat end. The bridge piece is pinned down to the lower half casing just outside the gland (see sketch), and the pin is adjusted to a known clearance, say $\frac{50}{1000}$ of an inch, from the top of the shaft, so that by applying this test arrangement at intervals any wear down of the rotor can be detected by means of "feelers" placed between the flat end of the pin and the upper surface of the rotor spindle (see sketch).



"Bridge" Gauge.

1. Rotor Spindle. 2. Steel Pin.

NOTE.—A record is kept of the original clearance between the gauge pin and the shaft at each end, and by reference to this record any subsequent wear down can be at once noted on testing.

NOTE.—From the foregoing descriptions of the two appliances, the "Bridge gauge" and "Finger plate," it will be obvious that the wear down and wear forward or aft of the rotor is under constant observation, and the risk of any serious accident, such as dummy ring stripping or rotor blade stripping, almost eliminated. The smallest possible change of rotor position can be accurately measured by the finely graded "feelers" used in testing the clearances shown by the gauges.

H.P. Drain to L.P.—The drain fitted at the after end of the H.P. turbine, and connected to either L.P. turbine and wet air pumps

is usually only opened when the turbines are stopped and shut when running. Sometimes, however, this drain is opened at regular intervals to clear the H.P. turbine of condensed water, but apparently this precaution is not absolutely necessary, as careful tests, conducted on behalf of the writer, prove that only water actually condensed in the drain pipe is blown out, as afterwards steam only is observed to blow through. From this it is obvious that most of the steam condensed into water during the adiabatic expansion of the steam in the H.P. turbine passes out with the exhaust steam to the L.P. turbines by way of the receiver pipes, and does not remain in the turbine casing as might have been supposed. This matter is one of considerable practical importance, affecting, as it does, the safe running of the turbines. As described elsewhere, the L.P. turbine drains are constantly open to the wet air pumps when running, but when the turbines are stopped the instructions marked up are: "When air pumps are stopped, switch all drains to bilges."

47. BLADING LIST.

(About 5,000 I.H.P. Speed, 16 knots.)

(NOTE.—The following figures are closely approximate.)

H.P. Rotor.

Diameter, 2 ft. 6 in.

Expansion.	No. of Blade Rows per Expansion.	Blade Heights.	No. of Blades per Row.
		In.	
1	12	$\frac{3}{4}$	500
2	12	$1\frac{1}{4}$	500
3	13	$2\frac{1}{8}$	500
4	13	$3\frac{1}{2}$	500

H.P. Cylinder.

1	12	$\frac{3}{4}$	390
2	12	$1\frac{1}{4}$	420
3	13	$2\frac{1}{8}$	450
4	13	$3\frac{1}{2}$	460

L.P. Rotor.

Diameter, 3 ft. 3 in.

Expansion.	No. of Blade Rows per Expansion.	Blade Heights.	No. of Blades per Row.
		In.	
1	5	$1\frac{1}{2}$	650
2	5	$2\frac{1}{8}$	650
3	5	3	650
4	5	$4\frac{1}{2}$	650
5	5	6	415
6	6	$7\frac{1}{2}$	415
7	6	$7\frac{1}{2}$	450
8	4	$7\frac{1}{2}$	390

L.P. Cylinder.

1	5	$1\frac{1}{2}$	500
2	5	$2\frac{1}{8}$	520
3	5	3	530
4	5	$4\frac{1}{2}$	590
5	5	6	460
6	6	$7\frac{1}{2}$	490
7	6	$7\frac{1}{2}$	520
8	7	$7\frac{1}{2}$	440

Astern Rotor.

Diameter, 2 ft. 5 in.

Expansion.	No. of Blade Rows per Expansion.	Blade Heights.	No. of Blades per Row.
		In.	
1	4	$\frac{1}{2}$	480
2	4	$\frac{11}{16}$	480
3	4	1	480
4	4	$1\frac{3}{8}$	480
5	4	2	480
6	6	$2\frac{3}{4}$	480
7	6	$2\frac{3}{4}$	470

Astern Cylinder.

1	4	$\frac{1}{2}$	380
2	4	$\frac{11}{16}$	385
3	4	1	390
4	4	$1\frac{3}{8}$	395
5	4	2	400
6	6	$2\frac{3}{4}$	410
7	6	$2\frac{3}{4}$	380

NOTE.—H.P. dummy contains 28 rings.
L.P. dummies contain 36 „
Astern „ „ 6 „ (radial fin type).

48. Finger-plate Readings and Turbine Pressures.

Type—Coasting Steamer

(16 knots; Horse-Power, 5,000).

No.	H.P. Initial Pressure.	P.L.P. Initial Pressure.	S.L.P. Initial Pressure.	Finger-plate Reading.			Date.
				H.P.	P.L.P.	S.L.P.	
				In.	In.	In.	
1	130 lbs.	0	5 in. vac.	.017	.027	.027	10/3/05
2	130 „	0	6 „ „	.017	.026	.028	21/3/05
3	115 „	4 in. vac.	9 „ „	.015	.021	.026	29/6/05
4	145 „	4 „ „	1 lb.	.017	.022	.032	22/9/06

49. Type—Fast Passenger Steamer.

Boiler steam	185 lbs.		
H.P. turbine initial pressure	150 „		
H.P. turbine terminal pressure	22 „		
	P.		S.
L.P. turbines initial pressure	21 lbs.		21 lbs.
L.P. turbines terminal pressure	25 in. vac.		25 in. vac.
Condenser vacuum	$27\frac{1}{4}$ in.		$27\frac{1}{4}$ in.
	P.L.P.	H.P.	S.L.P.
Revolutions per minute	480	474	467
	Forward.		Aft.
H.P. gland pressures	4 lbs.		3 lbs.
S.L.P. gland pressures	$\frac{1}{2}$ lb.		$\frac{1}{2}$ lb.
P.L.P. gland pressures	Steam just showing.		$\frac{1}{4}$ lb.
Speed	22.5 knots.		
Coal	Welsh.		
Oil temperature (beginning of run)	90 degs.		
Oil temperature (end of run)	125 „		
Trunk air pressure	3 in.		
Ash pit air pressure	$1\frac{1}{8}$ „		

NOTE.—The H.P. turbine drains to L.P. turbines (and therefore "wet" air pumps), is shut down five minutes after starting, and kept closed during the run. This practice is found to be satisfactory owing to the fact that most of the water condensed in the H.P. turbine is carried over with the steam into the two L.P. turbines (see page 212).

50. Reduced Speed Results.

	A.			B.		
Boiler -	185 lbs.			185 lbs.		
H.P. initial -	50 "			18 "		
H.P. terminal	2½ in. vacuum			17½ in. vacuum		
S.L.P. initial	3 "			20½ "		
P.L.P. initial	3 "			18 "		
S. astern	26 "			26½ "		
P. astern	26 "			27 "		
Condensers -	28¼ "			29 "		
Revolutions per minute	S.L.P. 325	H.P. 310	P.L.P. 320	S.L.P. 235	H.P. 225	P.L.P. 230
Speed -	16 knots.			11 knots.		
Dummy clearance (when running)	S.L.P. .025	H.P. .012	P.L.P. .026	S.L.P. .030	H.P. .014	P.L.P. .032

Oil to Glands.—In the new style of steam gland shown (page 124) an oil service is led to the four Ramsbottom rings which form the outer end of the packing. The oil supply (under pressure) is led in through the hollow stud (see sketch) connecting the hood and flange of the gland, and the lubrication thus arranged prevents undue wear of the rings, which usually gave trouble in the older type of packing, and necessitated regular overhaul and renewal at intervals of from four to eight months. At the same time it should be stated that the practice of admitting oil to the turbines is somewhat risky, and discounts to a considerable degree the advantage of this system of gland lubrication.

Dummy Casing Drains.—Small drain pipes fitted with brass cocks are fitted to the bottom of the ahead dummy casings to drain out the water which accumulates there, and which tends to produce corrosion between the brass strips and the cast iron of the casing or steel of the dummy as described elsewhere.

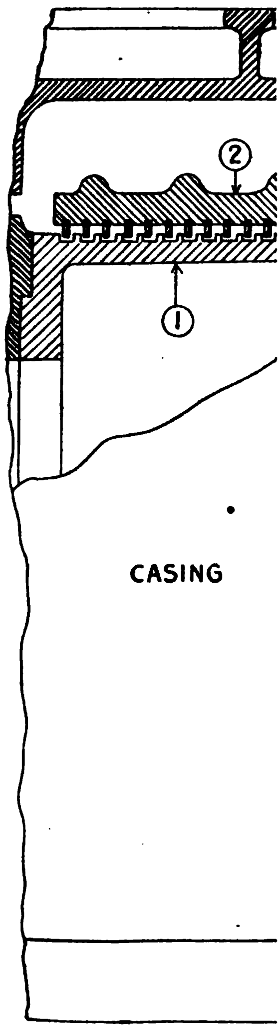
The H.P. dummy drain is led to the exhaust pipe connecting the H.P. and L.P. turbines, and the L.P. ahead drains are led to the wet air pump suction connections at the after end of the L.P. turbines.

Bilge connections are also fitted to drain off the water when the air pumps are stopped.

The reverse dummies drain off into the ahead casings by way of small holes bored through the reverse dummy casings.

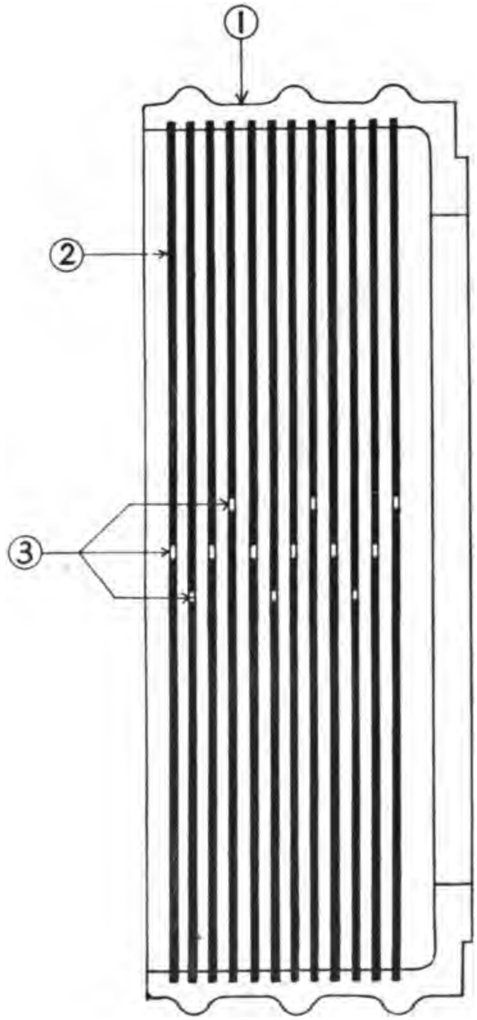
These drains are, as a rule, only opened when the turbines are stopped.

Draining of Dummies.—The dummies are sometimes drained by means of slots cut in the brass rings of the dummy casing. These slots are about ⅜ in. wide, and are only cut in the lower half case near the centre line—that is, at the lowest position. The condensed water drains off through the slots to the drain pipes fitted on the turbine casings.



(1) Ca
(2) Ro
(3) Br
(4) Mi

Micrometer Dummy Gauge.—The "finger-plate" method of testing the dummy clearance is now giving place to the micrometer gauge, which gives a much more reliable reading. A brass stop-piece



Casing Dummy (Lower Half).
Showing Method of Draining.
1. Casing. 2. Brass Rings. 3. Slots cut in Rings to form Waterways.

is screwed into the casing near the end of the dummy, and the gauge spindle, when set cold, and the dummy touching, bears up against this and indicates 0 on the wheel index. If then the wheel (7) is screwed

away clear of the handle, and a half-turn given to the spindle, the spindle end touches the actual end of the rotor, which, of course, is further aft than the stop.; if now the wheel is screwed *back* against the spindle collar (8) the index on the wheel circumference and the fixed pointer will read the dummy clearance in thousandths of an inch. The spindle end coming into actual contact with the rapidly revolving rotor is apt to wear away, which would result in a false indication being registered. The length of the spindle, however, can always be tested by means of the stop-piece (1), which acts as a check on the length, and indicates 0 if no wear has taken place; if, however, the spindle end has been gradually ground away, the wheel will indicate the amount when screwed back against the collar and the spindle end in contact with the stop. The amount of wear thus measured would require to be deducted from the reading of the gauge when in contact with the rotor as shown in the sketches.

EXAMPLE.—The micrometer gauge when tested against the "stop" registers $\frac{5}{1000}$ in., and when testing for dummy clearance shows $\frac{30}{1000}$ in.; determine the actual dummy clearance.

Then, $\frac{30}{1000}$ in. $-$ $\frac{5}{1000}$ in. $=$ $\frac{25}{1000}$ in. or .025 in. actual clearance.

To take a Reading.—(*a*) With micrometer gauge spindle (2) in contact with stop (1) see if wheel (7), when screwed back against collar, reads 0 on index (10).

(*b*) Screw wheel back clear of collar, and give spindle one-half turn by handle to bring spindle end up against end of rotor.

(*c*) Screw wheel forward again against collar, and note reading of index, which will then be the dummy clearance in thousandths of an inch.

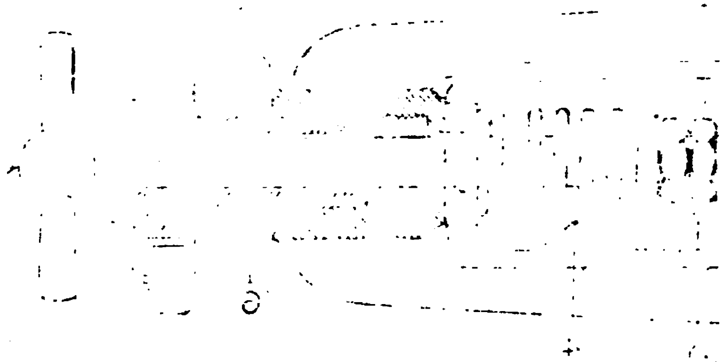
(*d*) Ease wheel again, and give spindle one-half turn to bring it back clear of rotor and into ordinary "out of gear" position against collar and stop.

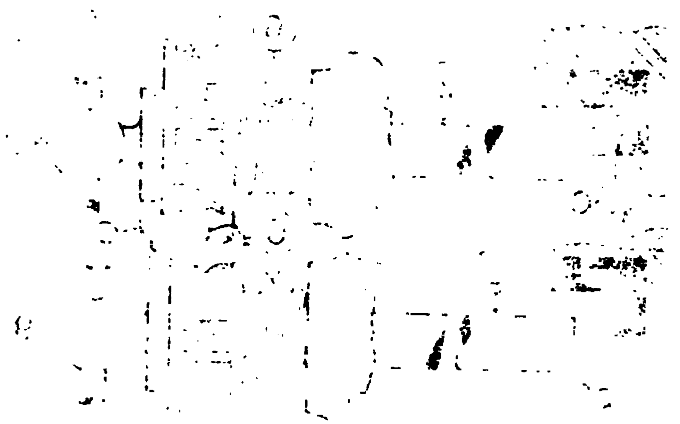
Expansion forward of Casing Feet.—As previously explained, the turbine casings are bolted down securely aft, and are allowed to expand forward when heated up; this is arranged by the bolt holes in the forward feet being larger than the bolts. In a case measured the movement was as follows :—

H.P. casing expanded forward - - - -	.055 in.
P.L.P. casing expanded forward - - -	.043 in.
S.L.P. casing expanded forward - - -	.045 in.

PROPELLERS.

Turbine Propeller Balance.—Turbine propellers are carefully balanced on knife edges, a spindle being passed through the hole in the boss for that purpose when shop testing, and the difference in weight made up or taken away on the astern or forward surface of the blade. The thrusting surfaces of the blades are then carefully





chipped and highly polished, the decided advantage obtained by this finish in reducing skin friction on the blades being now generally recognised.

Cones.—Turbine propellers are always fitted with long tapering cones to allow of the water from the blades having a clear run aft, and thus offering little or no resistance to the thrusting column of water.

Owing to the high revolutions necessary to obtain suitable turbine efficiency, the pitch of the propellers is small. An example will make this clear.

EXAMPLE.—Speed 17 knots, revolutions 400, and slip (assumed) 22 per cent.; find the required pitch of the turbine propellers.

$$\text{Then, } \frac{17 \times 6080}{400 \times 60 \times .78} = 5.5 \text{ feet Pitch.}$$

NOTE.—$100 - 22 = 78$ per cent., and $\frac{78}{100} = .78$ efficiency of advance.

The diameter of the propellers is also necessarily much less than usual, owing to limited space aft, and to obtain the required blade area a much larger ratio of blade surface to disc surface is developed. This fact chiefly explains the somewhat peculiar shape of blade met with in the majority of turbine propellers (see sketches), all of which are much broader in proportion to length than ordinary propeller blades.

It is sometimes found necessary during the period of the steamer's trial runs to alter the propeller blades, giving more or less surface as may be found by results to be beneficial, before the required speed can be obtained. This indicates that calculations relating to propeller design do not always give the results desired in practice, and emphasises the need for still more reliable information as to the true action of the marine screw propeller, and, if at all possible, for some standard method of accurate design, which will obviate the necessity for costly "trial and error" experiments with blades of various surface area and contour.

Pitch Ratio.—In ordinary marine reciprocating practice the propeller, the "pitch ratio," or pitch divided by the diameter, gives a number varying from 1 to 1.4, but in nearly all turbine practice the pitch ratio works out as a decimal figure, such as .8 or .9; in other words, the pitch is less than the diameter instead of being more.

EXAMPLE.—The pitch of an ordinary propeller is 15 ft. and the diameter 12 ft., find the "pitch ratio."

$$\text{Then, } 15 \div 12 = 1.25 \text{ Pitch Ratio.}$$

EXAMPLE.—The pitch of a turbine propeller is 8 ft. and the diameter 9 ft., find the "pitch ratio."

$$\text{Then, } 8 \div 9 = .88 \text{ Pitch Ratio.}$$

EXAMPLE.—The diameter of an ordinary propeller is 12 ft. and the projected blade area 34 sq. ft., find the "projected area ratio."

$$\text{Then, } 34 \div 12^2 \times .7854 = .3 \text{ Projected Area Ratio.}$$

EXAMPLE.—The diameter of a turbine propeller is 5 ft. and the projected blade area 12 sq. ft., find the "projected area ratio."

$$\text{Then, } 12 \div 5^2 \times .7854 = .6 \text{ Ratio.}$$

NOTE.—The *projected* blade area is the actual area of blades as seen from aft looking forward, and constitutes the effective "thrusting area" of the propeller. The "projected area ratio" is required in propeller design calculations, and varies from about .45 to .60 of the disc area.

Apparent Slip per cent.—In turbine steamers the apparent slip varies greatly with speed variation and weather conditions. A typical case is given below.

EXAMPLE.—At a speed of 21 knots the revolutions of the turbines of a channel steamer are 635 per minute; to find the apparent slip per cent. if the propeller pitch is 4 ft. 6 in.

$$\text{RULE.—} \frac{\text{Pitch} \times \text{Revolutions} \times 60}{6080} = \text{propeller speed.}$$

$$\text{Therefore, } \frac{4.5 \times 635 \times 60}{6080} = 28.2 \text{ knots.}$$

$$\text{Then, } 28.2 - 21 = 7.2 \text{ knots apparent slip.}$$

$$\text{And, } \frac{7.2 \times 100 \text{ per cent.}}{28.2} = 25.3 \text{ per cent. slip (apparent).}$$

Pitch Variation.—Turbine propeller blades generally present true screw surfaces, as no decided gain or advantage has been discovered hitherto by the adoption of pitch variation, either radially or peripherally, the all-important factors in propeller design consisting of the correct adjustment of pitch, diameter, and surface. At the same time, it should be mentioned that the propeller blades of the "Express" Cunarders were originally constructed with a peripheral pitch variation, the mean pitch being somewhere about 16 ft. This can be shown by the following: Speed 25 knots, revolutions 190, slip (assumed) 15 per cent.

$$\text{Then, Pitch} = \frac{25 \times 6080}{190 \times 60 \times .85} = 16 \text{ feet.}$$

Propeller Efficiency.—Above a certain revolution speed a propeller of given diameter, pitch ratio, and area ratio rapidly loses in efficiency as cavitation sets in and reduces the effective thrust. The slip ratio therefore increases, which produces a correspondingly reduced propeller efficiency : it is thus impossible to obtain a maximum propeller efficiency in combination with a maximum turbine efficiency. Propellers of comparatively small diameter also possess the disadvantage of losing in efficiency against head seas and head winds, which again results in increased slip ratio. The problem is then to combine both propeller and turbine speeds so as to give the highest possible *combined* efficiency. It will have been observed that, particularly in turbine steamers, the actual sea speed often falls away from

Turbine Propeller.

Pitch, 5 ft.; Diameter, 6 ft.; Pitch Ratio = .83.

(Khedive's Yacht.)

Messrs A. & J. Inglis & Co. Limited.

Propellers of Turbine Steamer.

(Khedive's Yacht.)

Messrs A. & J. Inglis & Co. Limited.

[*To face page* 220.

the trial trip speed, and the reason of this will perhaps be understood from the foregoing explanation.

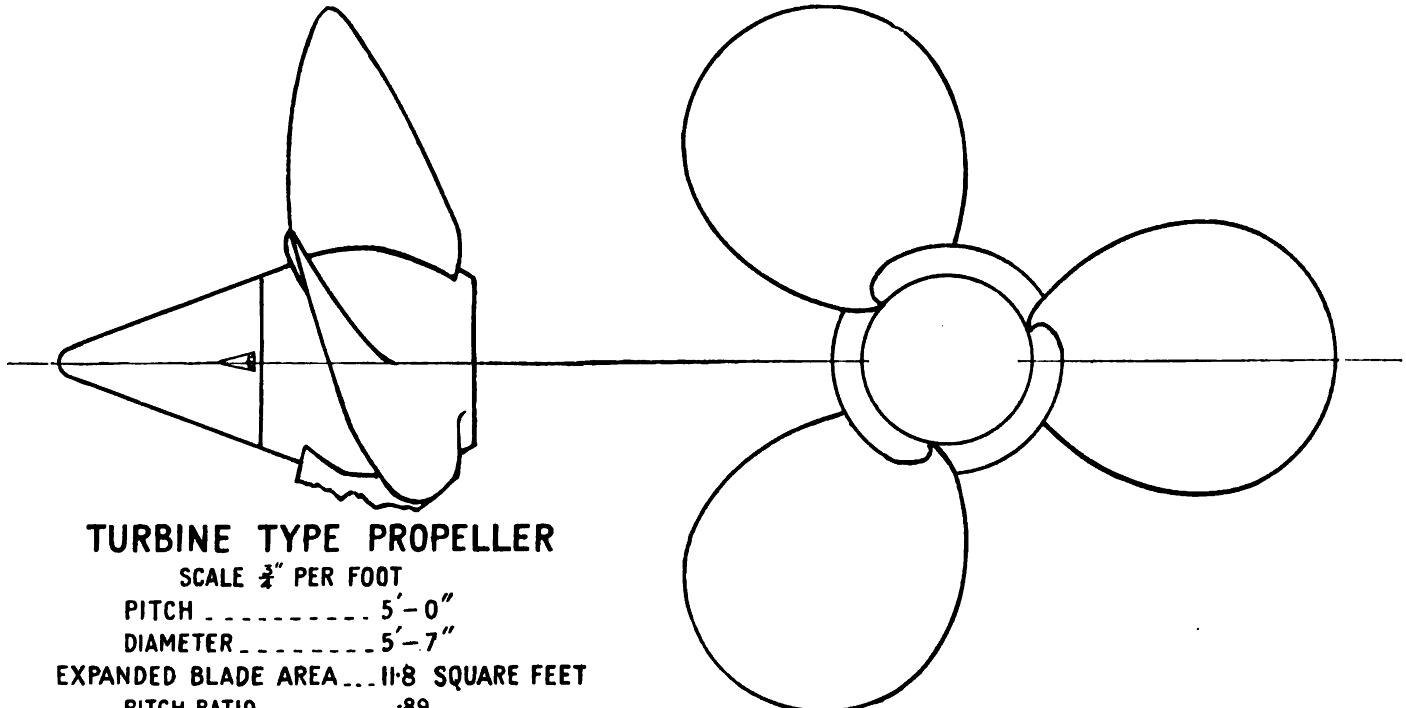
Cavitation.—Cavitation is caused by the ineffectiveness of the atmospheric pressure to press up the water at the back of the blades fast enough to allow of effective thrust. This usually occurs at high revolution speeds, and at high blade pressures per square inch.

Regarding cavitation and slip, Mr E. M. Speakman says :—

"Cavitation is partly the result of attempting to obtain too much work per square foot of blade area, and partly of excessive peripheral speeds. It has been found, by bitter experience occasionally, that there is a narrow limit to the tensional pressure possible on the water, per unit of projected area, beyond which the propeller efficiency drops very rapidly. This pressure is approximately from 10 to 12 lbs. per square inch at a depth of 12 inches below the surface, and to reduce the total thrust to this, sufficient blade area must be provided, which, in conjunction with certain practical proportions, necessitates a certain size of propeller, thereby limiting the revolutions.

"Friction and slip constitute the normal losses in all propellers, and augmented resistance must also be taken into account. This latter loss, however, is materially reduced with the smaller diameters of propeller found in turbine work. The percentage of slip has varied from 28 per cent. in H.M.S. 'Viper,' down to about 14 per cent. in the 'Viking'—channel steamers usually having about from 17 to 24 per cent. For large ocean-going vessels about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Comparing the 'Manxman' with the 'Ulster'—a Holyhead mail boat of similar speed and power—the total disc area of the twin propellers of the latter is 226 sq. ft. (for two twelve-foot diameter propellers), while that of the 'Manxman' is only about 80 sq. ft. If the thrust deduction is proportional to the absolute area of the disturbance of the steam lines at the stern, the effect on the 'Ulster' will be far greater than on the 'Manxman,' but this action is to some extent affected by the intensity over the disturbing area, which again is modified by the proximity of the propellers to the side of the vessel, this being less in turbine work. The disc area in H.M.S. 'Velox' is less than half that of the propellers in ordinary destroyers of the same power and speed. Cavitation is a preventable loss, and its presence on many vessels with insufficient blade area may be deduced from the falling off of the thrust curve and the rapid rise in the slip curve above a certain speed.

"From the analysis of numerous trials it appears that the pressure per square inch of projected area, when reduced to 12 in. immersion of tip, due to the effective thrust, is approximately 1 lb. for every 1,000 ft. per minute of circumferential velocity of blade tips. For a screw of a given pitch ratio, working at its maximum efficiency, this velocity should be proportional to the designed speed of the ship, and at full speed the pressures seem to have hitherto been about 5 lbs. per square inch for slow cargo vessels, from 6 to 7 lbs. for ocean-going mail steamers, and from 7.5 to 8.5 for cross channel steamers; in cruisers and battleships they vary from 8 to 10.5 lbs. in some recent notable instances, in a torpedo craft from 9 to



TURBINE TYPE PROPELLER

SCALE ¾" PER FOOT

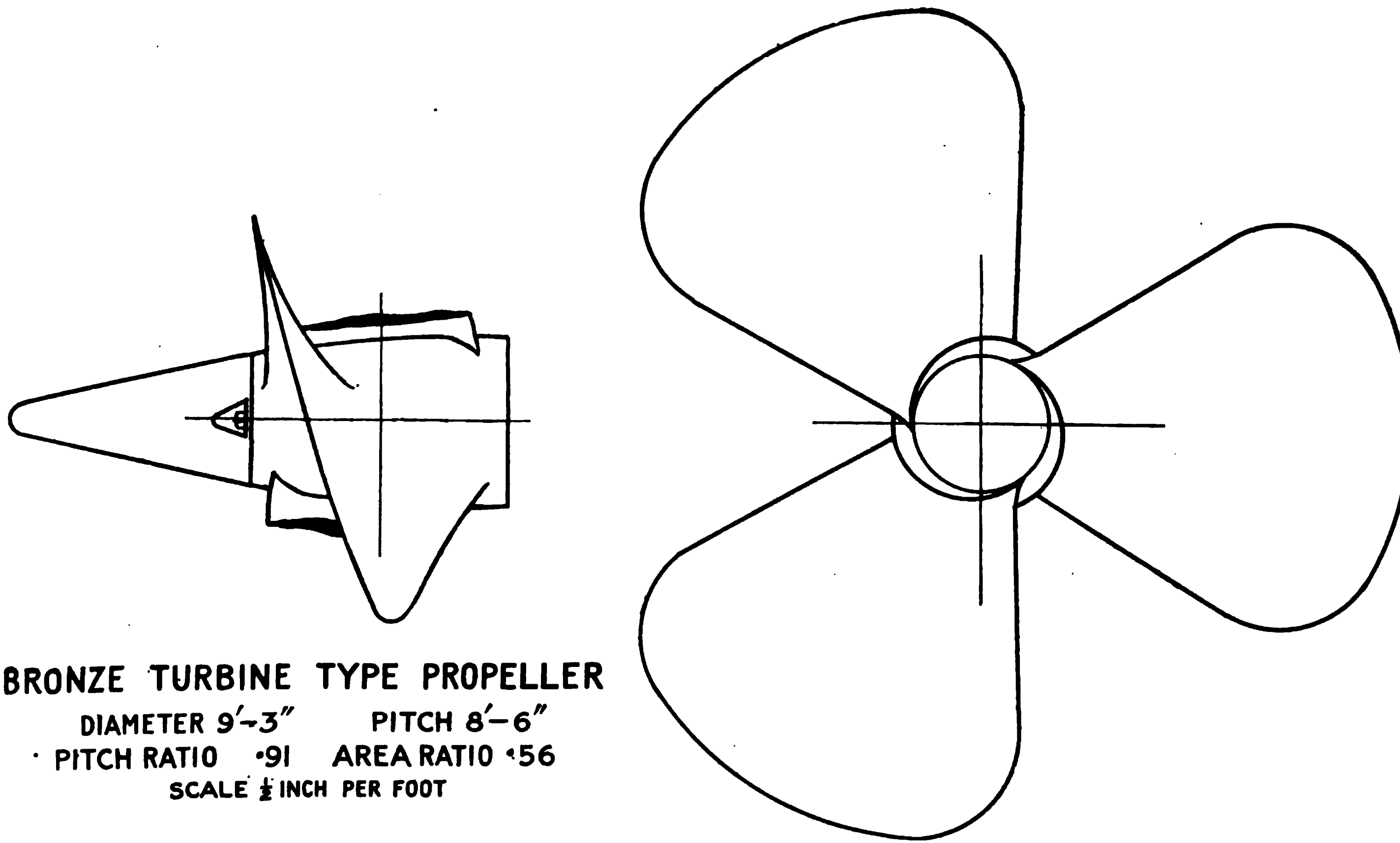
PITCH .... 5′–0″

DIAMETER .... 5′–7″

EXPANDED BLADE AREA ... 11·8 SQUARE FEET

PITCH RATIO .... ·89

AREA RATIO .... ·48



BRONZE TURBINE TYPE PROPELLER

DIAMETER 9′–3″ PITCH 8′–6″

PITCH RATIO ·91 AREA RATIO ·56

SCALE ½ INCH PER FOOT

11 lbs. At about 9 lbs., rather less possibly, the lowest suitable limit is reached for turbine screws; from 10 to 11 lbs. may be more usual in fast vessels, and though even from 12 to 14 lbs. has been known, pressures over about 11 lbs. always seem to be accompanied by low efficiencies. In large ocean-going vessels, which may be delayed by head winds and seas, a much lower designing pressure should be used, but in destroyers, as suggested above, something may be sacrificed at the maximum speed to obtain other advantages. The above pressure is worked out as mean pressure, as there exists no method of determining the local intensity per square inch, though the tendency of the distribution may be assumed in some cases. The only published results are in Barnaby's papers on the trials of

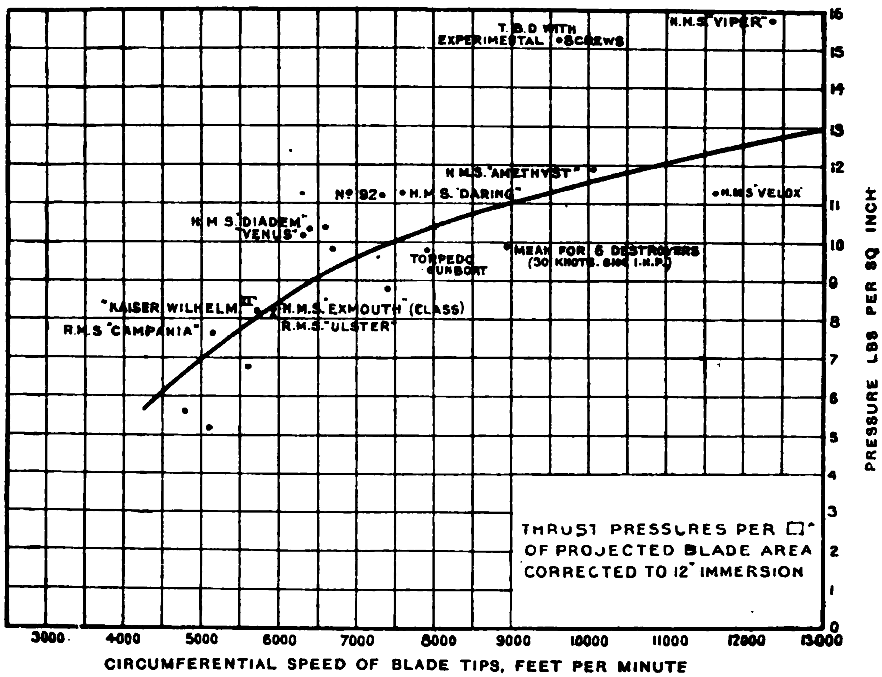


Fig. 1.

H.M.S. 'Daring,' but they are very inconclusive in many ways. Fig. 1 is submitted with much diffidence, merely as an illustration of the values of this limiting pressure.

"The maximum peripheral speed of tip ever used, I think, was 12,400 ft. per minute in H.M.S. 'Viper'; in H.M.S. 'Velox' it is about 11,650, and in the 'Londonderry' about 11,760, but many vessels have been below 9,000, which is quite normal for ordinary destroyer practice.

"The pitch ratio for turbine propellers has been purposely made considerably finer than usual. Thus, the pitch ratio for the 'Emerald' was about 0.6; in channel steamers and cruisers of from 18 to 25 knots it has varied from 0.8 to 1.0, and in torpedo craft from 1.0 in H.M.S. 'Velox,' to 1.35 in H.M.S. 'Viper,' and 1.6 in H.M.S. 'Cobra'; the latter vessels having 1, 2, and 3

screws per shaft respectively driven by identical turbines, the approximate revolutions at full speed being 900, 1,200, and 1,050 for 27, 36, and 31 knots respectively on trial.

"The percentage of slip has varied from 28 per cent. in H.M.S. 'Viper' down to about 14 per cent. in the 'Viking'—channel steamers usually having about from 17 to 24 per cent. For large ocean-going vessels about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Recent experiments in the model tank at Washington, D.C.,* seem to show that a symmetrical section will materially increase the pressure at which cavitation commences, and also demonstrate that in fine-pitch high-speed screws the back of the blade should receive almost as much attention as the face. This, as the author is well aware, is no new idea, but there have been repeated indications, especially in the trials of ordinary torpedo-boat destroyers, that while the gain may not be great, it is sufficient to merit attention. Mr Parsons has advocated a 10 per cent. reduction of pitch at the blade tip in order to avoid excessive local thrust, which might induce early cavitation, but there seems to be no advantage from departing from a true screw. The tendency of late years, in reciprocating-engine practice, has been to increase the ratio of projected to disc area from the .2 of Froude's classic screw, and the .22 to .26 of naval practice, to about .33; destroyer practice is included between this and .37, or even .4, at which point turbine practice may be said to commence. In this even from .5 to .56 has been used, but beyond, about .58 blade interference becomes excessive, and to obtain greater area a larger diameter must be used.

"The best form of blade is still undetermined. In the photo of the stern of the 'Lorena' will be seen the usual shape adopted, and experience seems to show that this almost circular shape, with the area disposed symmetrically on each side of the centre line, and with the generating line of the screw at right angles to the axis, gives as good results as any form.

"I find that the following formula will give the diameter of a turbine propeller with considerable accuracy when the effective thrust along the shaft is known, and this must be calculated in any case if the steam balance of the turbine is to be good:—

$$\text{Diameter of propeller in feet} = \sqrt{\frac{\text{effective thrust in lbs.}}{\text{coefficient}}} = \frac{\sqrt{T}}{C}.$$

"This coefficient has been deduced from the limiting pressure per square inch, and the ratio of projected to disc area, and is given in diagram form, in Fig. 2, where the full coefficient 400—900 is given in the left-hand scale, and values of C or $\sqrt{\text{Coefficient}}$ appear in the right-hand margin. The square root is only extracted for simplicity, whereby such coefficients as 30 are obtained for H.M.S. 'Viper'; while for the 'Manxman,' that for the centre screw is 26.4, and for the wing screws about 28.75. For large ocean-going vessels with lower designing pressures these values will be rather less, perhaps about 22.

"Compared with above values of C, reciprocating-engine practice gives such figures as:—R.M.S. 'Lucania,' 16.5; H.M.S. 'Diadem' class, 17.5; H.M.S. 'Exmouth' class, 19.0; and standard 30-knot destroyers of 6,100

* See D. W. Taylor's paper, American Society N. A., 1904.

H.P., 20.8; all of which have much lower ratios of projected to disc area, and therefore a larger diameter and smaller C for a given power. Regarding these figures, it must be understood that C is an approximation only, and, owing to

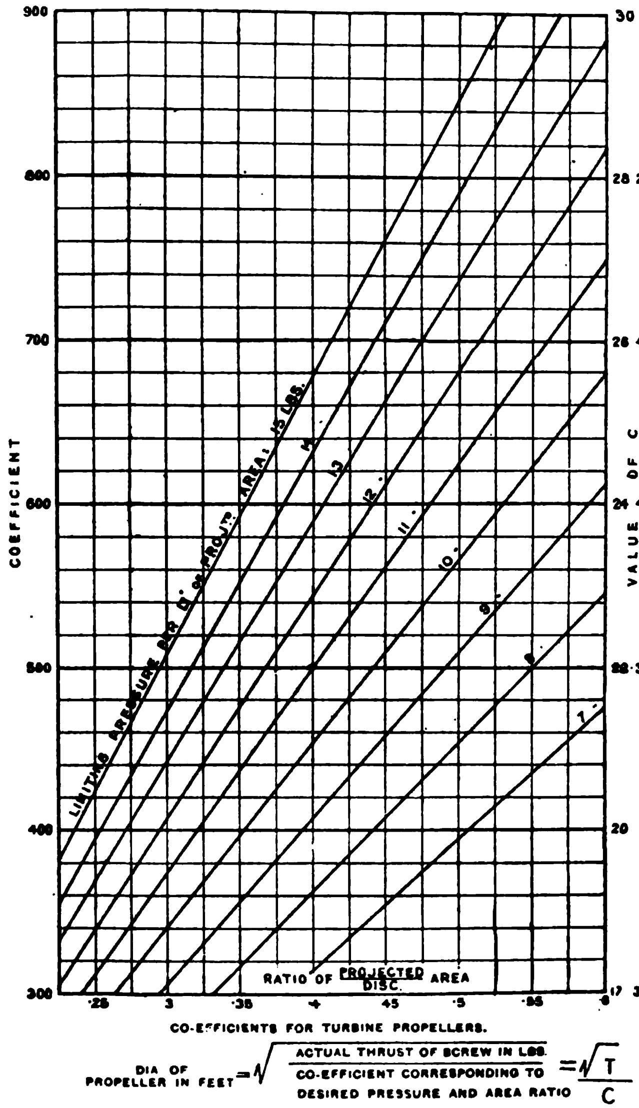


Fig. 2.—Propeller Calculations.

the difficulty of obtaining the actual power in each shaft in every case, cannot be considered as absolutely accurate. Probably, however, the error involved is under 2 per cent. It seems likely, to some at present undeterminable extent but within narrow limits for each class of vessel, that the propeller

efficiency is proportional to the coefficient C, and this seems to be borne out by the trials of the 'Manxman' and the 'Londonderry.'

"Effective thrust is a somewhat subtle subject, and our knowledge of propulsive efficiency is by no means what it ought to be. These considerations will undoubtedly be brought into far greater prominence in the near future, and it is by no means improbable that the Admiralty or certain private owners will require some definite standard in this, just as coal or steam consumption is regulated at present. The more propeller efficiency is studied and understood the greater will be the improvement in the design of turbine installations for marine work; the turbine itself is a comparatively secondary consideration, and while at present propeller dimensions for turbine steamers can be quite as closely determined as those for ordinary work, the exact proportions must necessarily largely remain subject to modification from actual experience.

"The following table gives a few propeller dimensions and the corresponding coefficients, which the author trusts will be of use in designing high-speed screws:—

Propeller Dimensions.

Vessel.	Type.	No. of Screws.	Diameter.	Pitch.	Pitch Ratio.	Speed of Tip.	C Approximate.
			ft. in.	ft. in.		ft. p. min.	
Turbinia -	Experimental	9 3	1-6 2-4	2-0 2-4	1.33 1.0	10,860 ...	28 * 31.3*
Viper - -	T. B. D.	8	3-4	4-0 fwd 4-6 aft.	1.2 1.35	12,350	30.1
Amethyst -	3rd class cruiser	3^{1}_{2}	6-6	6-6 5-10	1.0 .898	9,200 10,000	30.8
Manxman -	Cross-channel steamer	3^{1}_{2}	6-2 5-7	5-7 5-0	.906 .896	10,270 10,760	26.4 28.75
Londonderry	Cross-channel steamer	3^{1}_{2}	5-0	4-6	.9	10,550 11,810	30.8
Dieppe -	Cross-channel steamer	3	5-3	...	...	10,100 10,400	29
Carmania -	Atlantic mail	3	14-0	13-0	.928	8,125	21
Victorian -	Intermediate	3	8-30	...	...	7,150	24.65

* These C values are calculated from the same effective thrust in each case.

"While a general tendency has been very noticeable towards increasing the propeller diameter and reducing the revolutions, there will, of course, be some point, at present undetermined, at which the triple screws used in turbine work will be distinctly less efficient than ordinary twin screws. Very largely this is the case at present with triple screws driven by piston engines, on account of excessive thrust deduction and interference, but probably before this point is reached the weight of the turbines will have prevented its adoption.

"Having obtained the diameter of the propeller and the revolutions possible, the design of the turbine can then be undertaken, but for this no formulæ exist at present, such as are met with in reciprocating-engine practice."

Under actual sea-going conditions, the resistance to advancement is increased by wind, &c., so that to obtain the same advance more revolutions are required, but as this increase in revolutions often produces cavitation (the propellers and revolutions being already designed for the limiting conditions), so that the limit is exceeded, and loss of push or thrust results, conjointly with a high slip ratio; hence, 15 per cent. slip may be the trial result, but 30 per cent. slip the actual sea-going result.

From the foregoing it will be evident that increasing the revolutions and decreasing the pitch may not give identical results, although theoretically this should be the case. The phenomenon of cavitation upsets the calculations, and seriously affects the results, owing to reduced thrust efficiency. Generally speaking—

1. A high turbine efficiency means a low propeller efficiency.
2. A high propeller efficiency means a low turbine efficiency.

The best combined efficiency of turbine and propeller is what has to be aimed at, and this can only be obtained by sacrificing one or other or both of the two efficiencies referred to, so that a compromise is effected. Generally, the propeller efficiency is sacrificed, as the advantage of this results in a higher proportional turbine efficiency and economy.

Turbine and Propeller Efficiency Combined.—It is often advisable in turbine steamers to sacrifice propeller efficiency so as to obtain a high turbine efficiency. This accounts for the high slip ratio noticeable in many turbine steamers, as it is found better to drop some of the propeller efficiency to gain more turbine efficiency.

Hence with high revolutions the turbine efficiency will be good, but the propeller may possibly give a better result with less revolutions per minute. A compromise is thus effected to produce the highest possible *combined* efficiency of turbines and propellers. This then explains the high slip per cent. often recorded.

Speed of Rotation.—As mentioned elsewhere, a great deal of the economy of a turbine motor depends on the high rotational velocity of the rotor, which, of course, necessitates smaller propellers, so that if the revolution speed is lowered to allow of larger propellers being used, the diameter of the turbine rotor must be increased if the same steam velocity is to be maintained: this means, in consequence, a rapid increase in weight of the rotors and casings.

From the foregoing it will be obvious that, roughly, the most advantageous conditions for the turbine constitute the most disadvantageous conditions for the propeller, and *vice versâ*, but as very elaborate and costly tank experiments have been made, and are still being carried on, the design of the propeller is being much improved, and the efficiency increased for higher revolution speeds. Experience seems to indicate that the propeller will require to meet the turbine requirements, and not the reverse.

COMPARISON OF HIGH REVOLUTION AND LOW REVOLUTION SPEEDS ON TURBINE AND PROPELLER EFFICIENCIES.

HIGH REVOLUTIONS. (Say, 500 per min.)		LOW REVOLUTIONS. (Say, 150 per min.)	
Turbine Advantages.	Propeller Disadvantages.	Turbine Disadvantages.	Propeller Advantages.
1. Smaller turbines, therefore less weight of machinery.	1. Smaller propellers, therefore less holding power, or resistance against head winds and seas.	1. Larger turbines, therefore increased weight and space required for machinery.	1. Larger propellers giving increased blade surfaces, and more holding power.
2. Reduced blade clearance losses, as these vary inversely as revolutions squared.	2. Increased risk of cavitation due to increased resistance under severe sea-going conditions.	2. Largely increased blade clearance losses, as these vary inversely as revolutions squared.	2. Less pressure per square inch on blade surfaces, with correspondingly reduced risk of cavitation.
3. Better ratio of V.T. to V.S. to produce turbine efficiency.	3. Reduced margin for increase of speed if required as cavitation develops.	3. Great increase in centrifugal force due to increased weight of rotor.	3. Margin for increased speed without danger of cavitation developing.
		(The weight varies roughly as revolutions squared.)	

Head Wind or Sea.—With small propellers, the effect of a strong head wind or current in retarding the ship's speed is more apparent, and constitutes an appreciable disadvantage; in consequence of this, large deep-sea turbine steamers are much lower in revolution speed than cross-channel or river turbine steamers, the propellers being larger in proportion.

1. **Propellers and Power, &c.**—For a speed of 20 knots, and shaft horse-power of 17,500, the propellers, three in number, are 8 ft. diameter, 8 ft. pitch, and give an expanded blade surface of 28.4 sq. ft., the projected area being 26.4 sq. ft., and the mean revolutions at full speed being 366 per minute. The pitch ratio is therefore equal to 1 and the projected area ratio .52, as $26.4 \div 8^2 \times .7854 = .52$.

Then, $17500 \div 3 = 5834$ horse-power per screw, and assuming a propulsive efficiency of 52 per cent.

$$\text{Then, effective thrust pounds} = \frac{5834 \times 33000 \times \frac{53}{100}}{\frac{20 \times 6080}{60}} = 50338 \text{ lbs.}$$

Therefore, $50338 \div (26.4 \times 144) = 13.2$ lbs. pressure per sq. in. of projected blade surface.

Revolutions, Power, and Speed.

Revolutions.			Shaft Horse-Power.			Total.	Speed.	Slip.
P.	C.	S	P.	C.	S.			
188	214	184	630	1030	620	2280	11.5 knots	24°/₀
377	374	363	5945	6162	5366	17473	20.5 „	30°/₀

2. For a speed of 26 knots and shaft horse-power of 10,000 (each shaft) the propellers are each 10 ft. 6 in. diameter, and 11 ft. 4 in. pitch, and assuming a projected blade area ratio of .6:—

Then, projected area $= 10.5^2 \times .7854 \times .6 = 51.95$ sq. ft. The revolutions are 250 (designed).

Then, pitch ratio $= 11.33 \div 10.5 = 1.08$, and assuming a propulsive efficiency of 53 per cent.

$$\text{Then, effective thrust pounds} = \frac{10000 \times 33000 \times \frac{53}{100}}{\frac{26 \times 6080}{60}} = 66275 \text{ lbs.}$$

Therefore, $66275 \div (51.95 \times 144) = 8.8$ lbs. pressure per sq. in. of projected blade area.

3. For a speed of 23 knots and combined shaft horse-power of 10,000 (approximate) the propellers, three in number, are 6 ft. 9 in. diameter, and 6 ft. pitch, with an expanded blade area of 20.5 sq. ft., the projected blade area being 17.7 sq. ft., the mean revolutions at full power being 500 per minute.

Then, pitch ratio $= 6 \div 6.75 = .88$,
and projected area ratio $= 17.7 \div 6^2 \times .7854 = .62$.

Horse-power per shaft $= 10000 \div 3 = 3334$, and assuming a propulsive efficiency of 51 per cent.

$$\text{Then, effective thrust pounds} = \frac{3334 \times 33000 \times \frac{53}{100}}{\frac{23 \times 6080}{60}} = 24072 \text{ lbs.}$$

Therefore, $24072 \div (17.7 \times 144) = 9.4$ lbs. pressure per sq. in. of projected blade area.

THE CURTIS MARINE TURBINE.

The "Curtis" type turbine is now coming rapidly to the front as an efficient marine turbine, and recent trials have brought out the fact that the consumption per brake horse-power per hour is fairly constant at both low and high speeds. This result is chiefly due to the fact that the steam, being admitted by hand-controlled nozzles, one or more of these can be shut off as required for reduced speed, thus eliminating the wire-drawing losses which occur when the main stop-valve requires to be partly closed to reduce the steam flow and shaft speed.

The Curtis turbine is of the compound impulse type, the compounding being for pressure and velocity.

Generally described, the Curtis turbine may be said to represent a combination of the De-Laval and Parsons, as nozzles similar to the De-Laval type are arranged on the periphery of the wheels through which the steam is admitted to each stage, and fixed and moving blades similar to the Parsons type are arranged within the stages themselves.

Each stage consists of a set of expanding nozzles, and two or more rows of moving blades alternating with fixed blades. Between each stage is fixed a diaphragm with expanding nozzles. The total expansion of the steam, therefore, takes place in stages in the successive expanding nozzles, the kinetic energy developed at each expansion being absorbed as it passes through the successive moving and fixed blades of each stage.

It should be noticed that, being an impulse type turbine, the steam falls considerably in pressure in the nozzles of each stage before acting on the moving blades inside the stages, and this fall of pressure results in increase of velocity, the kinetic energy thus liberated from potential energy acting solely to produce increase of steam speed.

The revolutions can thus be kept down to bring out the efficiency of the propeller, which gives this type of turbine an advantage over the Parsons type, in which propeller efficiency is often sacrificed more or less for turbine efficiency. On referring to the trial performance of the three American scouts (page 250) it will be seen that the respective revolutions of the Curtis turbined "Salem" and Parsons turbined "Chester" were, at the 22.5 knots trial, 312.5 and 473.5, giving a difference of more than 150 revolutions per minute between the two.

It may be of interest to mention that a well-known Clyde firm have taken up and are at present experimenting with the Curtis type marine turbine with the express purpose of developing it for adaptability to moderate and low speed steamers, and for combination with the Parsons type.

The following descriptions are reprinted, with kind permission, from the pages of the *Journal of the American Society of Naval Engineers*, to the President and Council of which body the author's thanks are due:—

The general construction, as well as principal details, may be understood by the description following, and by reference to plate facing page 252, representing a marine turbine of about 3,500 B.H.P. running at 475 R.P.M.

The term "stage" implies the space within two diaphragms, containing rotating and stationary vanes and a set of nozzles. The casing A, made of cast iron, is built up of several sections of semi-cylinders bolted together, the upper half being secured to the lower by longitudinal flanges. Dished heads B, in one casting, are bolted to the ends of the cylinder and contain the shaft stuffing boxes C. The packing in these stuffing boxes consists of double sectional rings of pure carbon, the space between being steam packed. The main shaft D is of wrought steel and hollow, necessarily of a large diameter to secure rigidity. The rotors E are built up of cast hubs, forged rims, and boiler-plate sides riveted and screwed. The diaphragms F consist of cast or dished plates riveted to steel rings, the inner of which has a composition boss within which the rotor hub revolves, the outer one a dovetailed extension fitting a groove in the casing, which holds each diaphragm in position. They are made in one piece, as there exists no need for removal; the upper casing, however, can be lifted up by disconnecting the flanges. The blading for each stage is made up of stationary vanes G, and rotating vanes H, the former in grooves of sectional attachments screwed to the inside of the casing, the latter in grooves of the rotor rim. The vanes are not, as in most other turbines, inserted and fastened independently or individually, but are made up in blade segments from 10 to 12 in. long, consisting of a foundation ring, the vanes and a shroud band all cast together. When the vanes are short, they may be milled out of the solid, but are usually built up as described, the vane itself being extruded from blooms of a special composition. The bodies I, containing the nozzles, are secured to the casing at each stage, and consist of a casting with apertures separated by properly formed nickel-steel plates cast in the body. The steam inlet for the ahead stage is at K and for the backing stage at L, the exhaust nozzle for both being at M. Of the nozzles those in the first stage, both ahead and backing, are expanding, all the others are parallel. The rotor shaft rests in the bearings N and revolves in oil under pressure. Steam leakage outward in the stuffing boxes (or air leakage inwards) is brought to a minimum by connecting the annular spaces, between the carbon pack-

ing rings of the gland, to some intermediary stage where the pressure is only slightly above the atmosphere.

The steam velocities through the nozzles in the various stages are so arranged as to allot one-quarter of the total energy in the steam to the first stage and one-eighth to each of the other six. This is done to obviate an undesirably high shell pressure in the first stage, in which otherwise both the bursting and distorting stress would act detrimentally on the large shell area. Losses on account of the high frictional resistance, which occurs when a fast-rotating disc revolves in dense steam, are also, thereby, materially lessened. To illustrate the working conditions of a turbine of this kind the following data are given, which are supposed to answer to a turbine developing about 1,200 horse-power. The pressures put down in the third column, which are the stage pressures, convey clearly the situation. For any other turbine of different power, but working under similar conditions, the corresponding pressures will, of course, be nearly the same.

TABLE I.

Stage.	Absolute Inlet Pressure, lbs. per square inch.	Absolute Shell Pressure, lbs. per square inch.	Throat Pressure in Nozzle, lbs. per square inch.	Square inch of Throat required, one Nozzle.	Absolute Nozzle end Pressure in lbs. per square inch.	Square inches of Nozzle end required, one Nozzle.
1	265.0	79.0	152.9	1.339	95.7	1.5048
2	79.0	41.7	45.58	4.24	45.58	4.24
3	41.7	21.2	24.0	7.8	24.0	7.8
4	21.2	10.4	12.23	14.84	12.23	14.84
5	10.4	4.9	6.0	29.25	6.0	29.25
6	4.9	2.2	2.83	60.66	2.83	60.66
7	2.2	1.0	1.27	135.0	1.27	135.0

The first-stage nozzles have each a separately operated disc valve for shutting off or opening to steam. All of the other stages are furnished with slide valves by means of which a certain number of their nozzles can be closed.

The steam flow through the turbine may accordingly be regulated to correspond with an amount of steam required for any desired horse-power, and the function of these valves thus bear the same relation to an economical variation of the power as do cruising turbines with Parsons' system. Regulating valves as described are necessary on turbines for naval ships cruising the greater part of their time at low power. For ordinary service the regulation may be accomplished by the throttle, and those valves are, therefore, limited on mercantile turbines.

Besides the independently operated stage valves pressure gauges are located on the shell to ascertain the pressure within each stage. A drain valve connects each stage in such a way as to lead the drain from a

preceding stage direct into the next following and successively into the exhaust cavity, at which point a connection is made with the condenser.

Arrangement.—Machinery installations with Curtis turbines for propelling purposes of large power, as mentioned before, are arranged for two shafts in the ordinary way (see diagram facing page 248). The disposition of the auxiliary machinery is, in general, also similar to that of arrangements with reciprocating engines. Two separate steam inlets are, however, required, one for ahead the other for backing, the pipes of which, with their throttle valves, connect to the nozzle bowls of the first stage. As there exists only slight end thrust on the rotor from uneven steam balance on the vanes within the cylinder, nearly all of the propeller thrust must be provided for in the usual manner in form of a thrust block, which is placed either forward or aft. The vane clearances being comparatively liberal all around, and there being an absence of dummy pistons with their close settings, renders micrometer adjustments in the thrust block entirely unnecessary. Gauges are, however, provided at the front and back of the turbine to enable the precise location of the rotor, with reference to the stationary parts, being determined.

Design.—The real difficulty when starting out in designing a steam turbine centres around the ability to determine the efficiency, or, in other words, to closely approximate the pounds of steam needed per horse-power for given conditions of pressure at entrance and exhaust and defined stipulations as to the quality of the steam. To do this in any turbine, and especially in turbines of the impulse type, like Curtis or De-Laval, it is necessary to base one's assumptions on data secured in tests or by actual experiments. Such tests have been performed most carefully with the present seven-stage Curtis marine type. An average result of such tests justifies us to assume a steam consumption of 14 lbs. per brake horse-power, using dry saturated steam at an initial pressure of 265 lbs. absolute and a vacuum of 28 in., the rotor vane speed at the same time to be approximately between 180 and 190 ft. per second and the allotment of power as previously stated among the seven stages.

With the use of superheated steam, for which this type of turbine is particularly well adapted, the consumption will undoubtedly be considerably lowered, down to 13 lbs. or better, which has been amply verified in various stationary turbine plants.

The total available heat per pound in steam expanding adiabatically between 265 lbs. and 1 lb. absolute, determined by the formula

$$H_1 - H_2 = Q_1 - Q_2 + H_v - T_2 \,(E_{q1} + E_v - E_{q2})$$

we find =

$$380.2 - 70.0 + 825.6 - 563.3\,(.5728 + .952 - .1329) = 352.2 \text{ B.T.U.}$$

If all of the heat contained in the steam could be converted into work on the shaft the consumption of the ideal turbine would be

$$\frac{1{,}980{,}000}{352.2 \times 778} = 7.21 \text{ lbs. per horse-power hour.}$$

The actual turbine, on the basis of 14.0 lbs. steam per horse-power hour, will accordingly give an efficiency $= \frac{7.21}{14.0} = 51.52$ per cent.

The losses in an impulse turbine may be attributed to the following causes:—

1. Steam friction in nozzles.
2. Steam friction in vanes.
3. Steam shock on the vanes caused by deviation in the relative entrance velocities and by starting the quiescent steam between the vanes when passing nozzles.
4. Resistance to the revolving surfaces in steam of more or less density.
5. Radiation of heat from the cylinder, spreading and leakage of steam between vanes, nozzles, and shaft bushings.
6. Loss of energy due to the exit velocity.
7. Mechanical losses, such as friction of journals and stuffing boxes.

The Nozzle.—Weight, space, and propeller speed renders it necessary, in marine turbine installations, to limit the rotor diameter as well as its speed of revolutions, and incidentally to use low steam speeds. These are obtained by arranging the turbine for a comparatively large number of "expansions" or stages and by using appropriate types of nozzles.

The general principles of steam expansion and the velocity attained by the jet in different nozzles has been written up extensively in various treatises and does not come within the scope of this paper. Suffice it to say, therefore, that the jet velocity depends essentially upon the "ratio of expansion," or on the proportions observed in the areas of throat and outlet end of the nozzle and upon the initial and terminal pressure at the orifice and outlet; moreover, that a definite relation always exists between the initial pressure and that of the throat, when the orifice is well rounded. Thus if P signifies the initial pressure, the throat pressure will be .577 P if the outlet pressure is equal to this or any other pressure below .577 P down to a perfect vacuum. This occurs in either a parallel or an expanding nozzle.

Action.—The Curtis marine turbine is of the compound impulse type. The steam expands in seven sets of nozzles or pressure stages successively from the initial pressure to that of the exhaust. The first pressure stage, as has been mentioned, develops one-fourth the total energy, the other six each one-eighth. The jet energy is transformed into work by impulse on the moving discs, which, according to the jet velocity used, are arranged in from three to four velocity stages. Good efficiency demands that certain proportions of steam and bucket velocity be observed throughout the turbine.

The steam action is essentially as follows: After expansion in the nozzles of the first stage the steam issues in solid jets against the first row of moving buckets, which absorb a part of the jet energy,

and, after passing through said buckets, meets the first row of stationary vanes. The purpose of these vanes is to guide the steam into the second row of moving buckets, which, in their turn, take up another portion of the kinetic energy still possessed by the fast flowing steam, at the same time diverting it into the second row of stationary buckets, which deflect the steam on the third row of moving buckets, where again more of the energy is taken up. In the first stage, where the initial nozzle velocity is considerably higher than in any

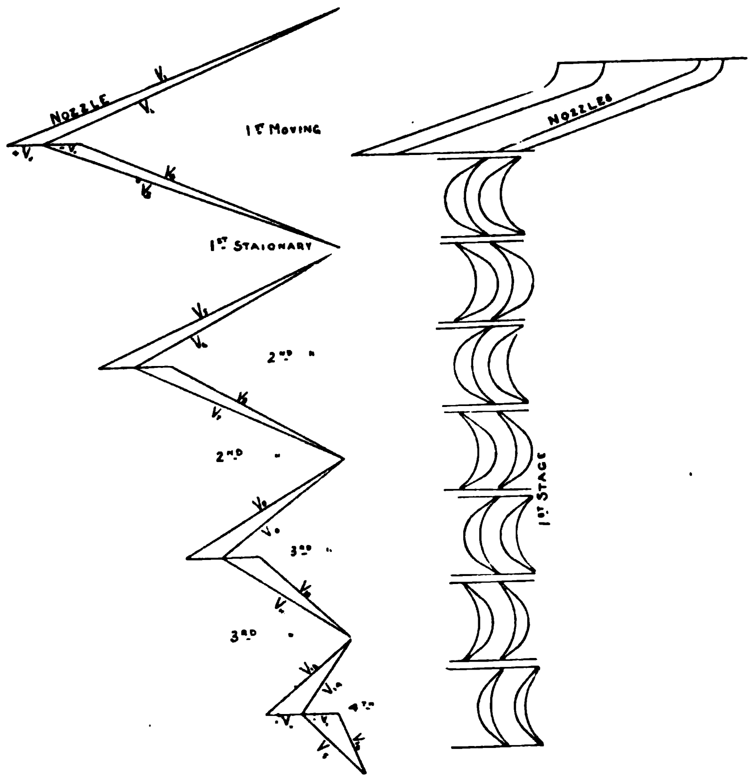


Fig. 5.
Velocity Diagram, "Curtis Turbine," first stage.

subsequent stage, this operation is again repeated in a third stationary row and a fourth moving row, the remaining energy after that being too small to warrant additional buckets. The pressure at the outlet end of the nozzles is brought down to the pressure within each stage by slight expansion in the various rows of buckets, the volume of the steam corresponding to successive pressures. However, due to the fact that the velocity of the steam is gradually diminished by the continuous absorption of energy, the passages traversed by the steam

must be enlarged. This is provided for by lengthening the buckets as well as by increasing the vane angles in each succeeding row.

After leaving the last row of moving buckets in each stage the steam attains partial rest before it enters the nozzles of the next stage, in a manner similar to that which occurs between each "expansion" of a Parsons turbine. On entering the nozzles of the second stage the steam again expands, whereby new velocity is given, and now acts in the various rows of buckets of that stage exactly as it did in the first stage, and so on right through all of the seven stages of the turbine. There is this difference, however, that, owing to the nozzle velocity being very much less in the stages succeeding the first, three moving and two stationary rows will suffice there instead of respectively four and three of the first stage. The number and size of the nozzles in the different stages must obviously conform to the velocity and the volume of the steam as a result of expansion through the turbine. Due to this fact we find the nozzles circumscribing only a small arc in the first stage, gradually increasing in the following, until, in the last stage, the entire circle is completely filled with nozzles. This latter condition, however, is governed wholly by the power in comparison with the rotor diameter.

Steam velocities through the buckets of the first stage relatively to a fixed vane speed are shown diagrammatically in Fig. 5. No account has been taken of velocity increase as a result of expansion in the buckets in this diagram, which must be done in figuring vane dimensions.

Calculation.—In a steam turbine, the potential energy of the admitted steam is converted into kinetic energy of the steam jet and the kinetic energy of the steam jet into impulse force on the revolving wheel. This double transformation can take place either simultaneously in the same wheel (reaction turbine), or successively in stationary nozzles and revolving wheels (action or impulse turbine).

In the reaction turbine, steam enters the revolving wheel at admission pressure at moderate velocity, expands in the wheel and leaves the wheel at the velocity of expansion relatively to the wheel, that is, in a vane completely reversing the jet, at an absolute velocity equal to expansion velocity minus wheel velocity. To take the full energy out of the steam, it must leave the wheel at zero absolute velocity; that is, the wheel velocity must equal the velocity of the steam jet.

In the impulse turbine, the steam expands in a separate nozzle, strikes the rotating wheel at expansion pressure and expansion velocity, and rebounds therefrom, as an elastic body, with the same velocity, minus the friction loss, as that with which it strikes, hence losing velocity by an amount equal to twice the wheel velocity. To take the full energy out of the steam, the wheel velocity must therefore equal one-half the velocity of the steam jet.

In an impulse turbine, therefore, such as the Curtis turbine, with a given pressure range per wheel, the wheel velocity at best efficiency is only about 50 per cent. of what it is in a reaction turbine, and $\frac{1}{\sqrt{2}} = 70$ per cent. of what it is in a combination turbine alternating between impulse and reaction effect, such as the Parsons turbine.

Since the most difficult condition of turbine design is the high rotor velocity required to utilise the high steam velocity, the impulse turbine is theoretically superior. It is not possible to get efficiency with a single turbine wheel of the reaction type, while with a single wheel of the impulse type good efficiencies have been reached by using extremely high peripheral speeds (De-Laval turbine). In general, however, while hydraulic turbines are always designed with a single wheel, in the steam turbine it is not possible to make effective use of the total energy of the steam by a single wheel, but a number of wheels in series have to be used. That is, the rotor velocity is reduced by subdividing the total pressure range into a number of successive stages. In the combination impulse and reaction turbine of Parsons, each expansion or pressure stage requires a revolving wheel and a stationary guide wheel, the steam acting by impulse at entrance and by reaction at the leaving edge.

In the impulse turbine, in which the expansion of the steam is carried out in nozzles separately from the vane system, a further effective means of speed reduction is presented by the use of several velocity steps in each stage; that is, by imparting the velocity of the current of steam to a number of successive wheels composing a single revolving disc, with stationary intermediate guide wheels.

Since in a revolving wheel of an impulse turbine the steam velocity is reduced by an amount equal to twice the wheel velocity for maximum efficiency, a single wheel per stage must revolve at one-half the steam velocity, two wheels per stage at one-quarter steam velocity, &c. The use of several wheels per stage, therefore, is a more effective means of reducing the rotor speed—or inversely, at given rotor speed, reducing the total number of revolving wheels—than the use of several expansion steps. A two-wheel stage can take care of twice the steam velocity (that is, four times the steam energy) of a single-wheel stage, and therefore replaces four single-wheel stages; or, in other words, the speed reduction of the rotor is proportional to the number of wheels per stage, that is, the number of velocity steps; on the other hand, it is proportional to the square root of the number of stages, that is, number of pressure steps.

The simultaneous use of pressure steps, or expansion stages, and velocity steps, or number of wheels per stage, therefore, leads to a construction requiring a comparatively small total number of wheels, as carried out in the Curtis type of turbine.

The following items are especially interesting with respect to the impulse type of turbine in review with the compound reaction type:—

1. For a given bucket speed and pressure range the number of vane rows is considerably less in the former than in the latter.

2. The steam expansion being executed in nozzles renders possible the use of high initial pressures, as well as superheat, without affecting detrimentally the turbine cylinder from excessive stress, or the vane system from distortion incidental to superheat.

3. Aside from throttle-valve regulation the steam supply may be varied, to suit any degree of power to be developed by the turbine, by shut-off valves on the nozzles, eliminating thereby the necessity for separate cruising turbines as now usually arranged for in warships.

4. Large clearances are used in Curtis marine turbine at both side and end of blades, thus minimising the danger of fouling of buckets from displacement of shaft or vibratory influences.

5. Balance or dummy pistons, to equalise end pressure, do not exist in the impulse marine turbine, and a source of loss, as well as the necessity for delicate adjustment, is eliminated.

6. For the same total power propeller diameters are larger in twin-screw ships fitted with Curtis turbines than in vessels of triple or quadruple screws with Parsons. The capacity for manœuvring is increased by using larger propellers and by making available all of the blade area against the two-thirds to one-half provided for in turbine ships of latter screw arrangement. Backing turbines are, however, fitted on each shaft in quadruple screw arrangements of naval ships.

Appended will be found a description of the Curtis turbines at present under construction at the Fore River Shipbuilding Company for the Japanese cruiser "Ibuki" of 14,600 tons displacement:—

Curtis Turbines of Japanese Cruiser "Ibuki."

The motive power of the "Ibuki" consists of two turbines, which are designed to develop a normal horse-power of 24,000, sufficient to drive the "Ibuki" at 22 knots speed. They are intended, however, to develop a maximum overload horse-power of 27,000, which should suffice to give a speed of nearly 23 knots. The rotor is 144 in. in diameter. The casing has an outside diameter of 14 ft., and a length over all of 17 ft. The weight of the two turbines together is 360 tons.

Each turbine consists of a cast-iron cylindrical casing divided by dished diaphragms into a series of separate compartments. In each compartment or "stage" there is a separate wheel, which carries on its periphery three rows of moving buckets (for reasons later described the first wheel has four rows). The wheels are all mounted on a hollow steel shaft carried by two bearings. Where the shaft passes through the diaphragms they are provided with bronze bushings having a small clearance, thus preventing appreciable steam leakage from one stage to the other. Where the shaft passes out through the ends of the casing it is provided with carbon stuffing boxes, which prevent steam leaking out at the ahead end, or air leaking in at the back end where a vacuum exists.

The stuffing boxes are supplied with steam in the space between the carbon packing to prevent air leaking in and lowering the vacuum. They are also drained to the fourth stage shell.

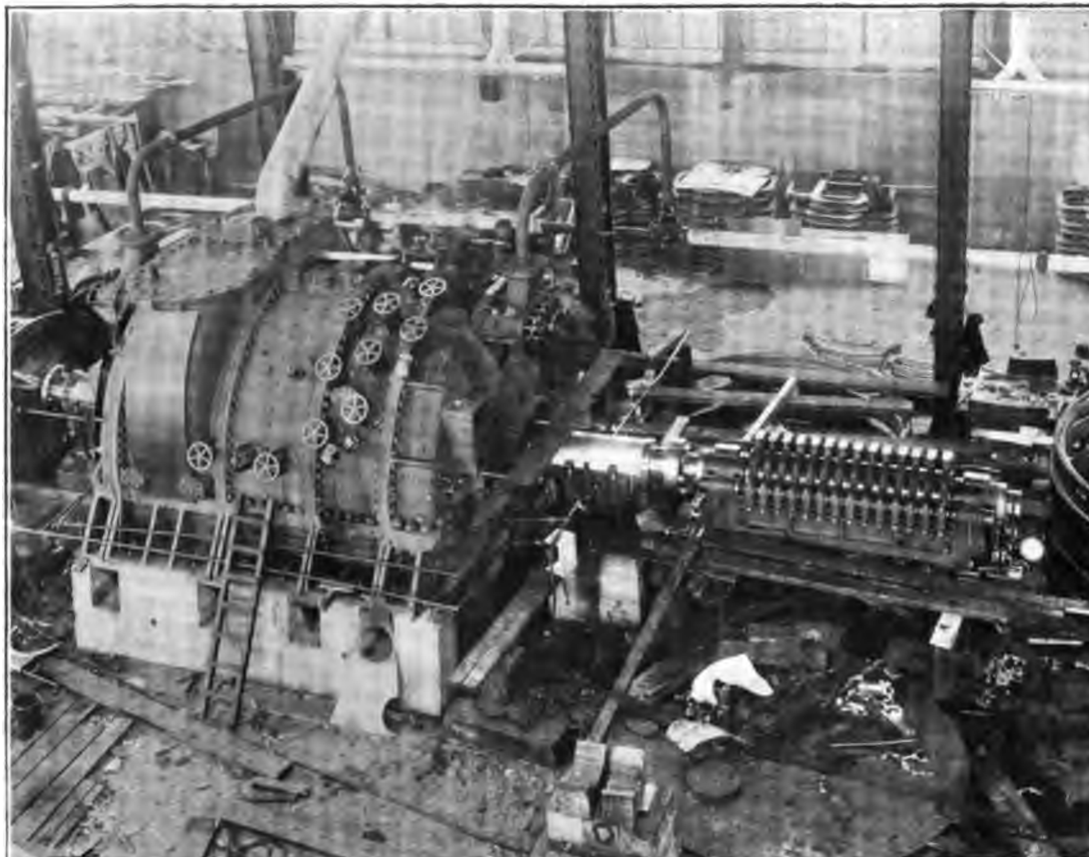
Cast-steel steam chests for ahead and astern running are attached to the front and back casing heads, and are flanged for the main steam pipes. The nozzles for each stage are bolted to the diaphragms, the diaphragms having steam-port openings cast in them to allow the steam to pass through to the nozzles.

Manœuvring is accomplished by means of two lever-operated balanced throttle valves, each taking steam from the main steam pipe, one delivering to the ahead steam chest and the other to the astern steam chest. There are seven ahead wheels and two reverse wheels. The reverse wheels are mounted in the after end of the casing, and under ordinary ahead running they are in a vacuum and, therefore, do not waste power by steam friction. They are similar to the ahead wheels, except that the blades are reversed. To reverse when going ahead, the ahead-throttle valve is shut and the reverse-throttle valve opened, which is easily and quickly accomplished by the operating levers of the two throttle valves.

Drain pipes are provided, connecting each stage with the next, so that the condensed steam in any stage will pass to the next one of lower pressure, and there give up a part of its heat to do useful work. The exhaust chamber drains to the condenser and the discharge is assisted by a small steam ejector. A regular marine thrust bearing is attached to the forward end of the turbine shaft. In addition to taking the propeller thrust, this bearing also maintains the proper axial position of the rotor, so that the axial clearance of the blades is correct. This clearance is one-tenth of an inch on the first wheel and increases to one-quarter of an inch on the seventh wheel. The thrust is put at the forward end, so that any unequal expansion of the shaft and casing will be allowed for at the aft end, where the clearance is largest. This axial clearance is very ample to allow for all unequal heat expansion that may occur and any mechanical irregularities, and leave sufficient leeway for adjustment.

To allow for the increased volume of the steam as it expands in passing from stage to stage at lowering pressures, the lengths of the blades are increased and also the arc of the nozzles is increased, thus giving greater area of passage in each succeeding stage. Also, in any one stage, the blade lengths are increased in each succeeding row, because the velocity falls as the steam passes from row to row, although it is at practically constant pressure throughout the stage.

In order to keep the pressure in the shell as low as possible, the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the first stage and one-eighth in each of the other stages. This requires the first-stage nozzles to be of the expanding type, but all the other nozzles are of the parallel-flow type. Also, the first-stage wheel is provided with four rows



Shop Test of 12,000 H.P. Curtis Turbine of Japanese Cruiser "Ibuki."
Constructed by Fore River Shipbuilding Co., Mass., U.S.A.

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of buckets instead of three, as on the other wheels, since the greater energy drop produces greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. This arrangement makes all the ahead wheels except the first operate under eight-stage conditions. The principal advantages of the Curtis design of marine steam turbine are as follows: Small number of blades; large clearance around blades; strong mechanical construction of blading; economy at reduced speed, without cruising turbines; interior of shell not subjected to full steam pressure; low revolutions for given horse-power; absence of dummy pistons and packing.

The small number, large clearance and strong construction of the blades make blade stripping practically impossible, and no case has occurred.

By the use of valves on the nozzle openings of the diaphragms, the proper steam-pressure distribution can be maintained at reduced steam flow, thus keeping up the economy at low speed of vessel, except, of course, for the unavoidable loss due to lower revolutions and dispensing with cruising turbines.

Full steam pressure comes on the steam chest only, which is a comparatively small steel casting. The greatest pressure in the turbine shell is less than one-third the working steam pressure. This permits high steam pressure to be used, and large turbine diameter in comparison to the power. It also reduces expansion difficulties.

The comparatively low revolutions permissible for a given power without sacrifice of economy or excessive weight allows the twin-screw arrangement to be used instead of three or four screws. Also other conditions being the same, lower revolutions will give a higher efficiency of the propeller. Low revolutions also permit the use of turbines in comparatively (for turbine vessels) low-speed vessels.

Absence of dummy pistons and their packing eliminates the leakage of high-pressure steam and makes the economy independent of any adjustments, so that the initial economy will be maintained continuously and will not be affected by any wear. — *Scientific American.*

Trials of U.S. Scout "Salem."

The following notes, referring to the trials of the U.S. cruiser scout "Salem" and of the Curtis type turbine, are reprinted from a booklet issued by the Fore River Shipbuilding Company, Quincy, Massachusetts, U.S.A., with the kind permission of the firm mentioned:—

"The change from kinetic energy to work is achieved by the 'impulse' due to jets of steam acting upon blades formed on 'wheels' mounted on the shaft to be rotated. The steam expands in a number of sets of nozzles or pressure 'stages' successively from the high pressure to the exhaust end of the turbine. Thus, after expanding in the nozzles of the first 'stage,' the steam issues in jets against the first row of buckets on the rotating wheel, a large part of

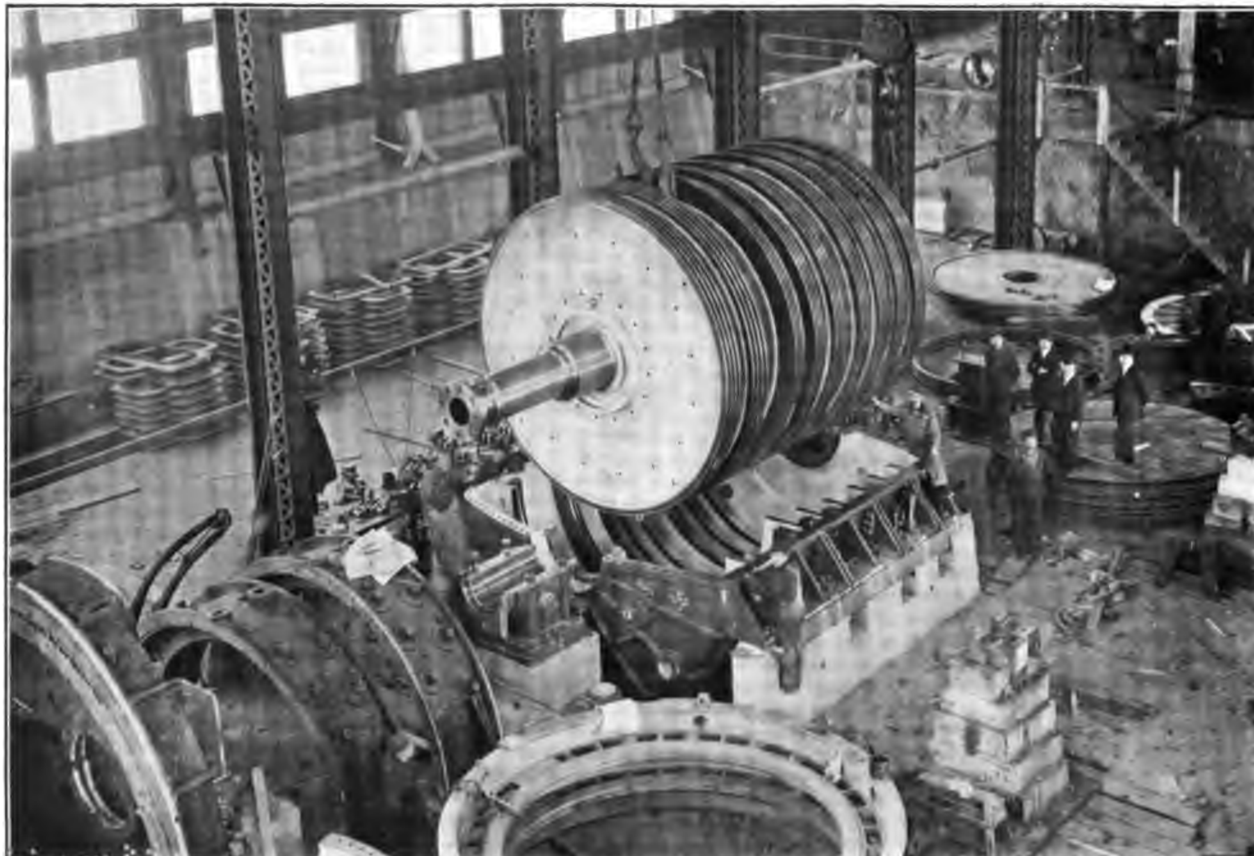
the energy being absorbed. It then flows to a row of stationary vanes, which guide the steam into a second row of moving buckets. These may be followed by a second set of fixed vanes and a third set of moving ones, after which the steam leaves the 'stage,' as it is called, through a second set of nozzles, where further expansion takes place, again generating velocity. From these nozzles it flows once more in sinuous fashion through successive sets of moving and fixed blades, and thence to other 'stages.' The important point to note is that expansion of the steam takes place only in the nozzle, and not in either the fixed or moving blades. Hence the pressure of the steam does not alter between one set of nozzles and the next. At the low-pressure end the nozzles cover the whole periphery of the wheel, but at the high-pressure end they extend only over an arc often not more than one-eighth of the whole circumference. It is thus possible to reduce the power of the turbine by cutting out a proportion of the total number of nozzles, instead of by reducing the pressure of the steam supplied by throttling it at the valve. Thus, whereas in the Parsons system cruising turbines are fitted to attain reasonable economy at low speeds, they are unnecessary with the Curtis system.

"The Curtis turbine occupies a middle position between the high-speed Parsons and the low-speed reciprocating engine; and, because of the moderate speed of revolution, and the fact that the power can be developed upon two instead of four shafts, it has become possible to secure a high propeller efficiency."

. . . .

"The propulsive efficiency of the 'Salem' rose from 55 per cent. at 12 knots to a maximum of 62.8 per cent. at the contract speed of 24 knots, then fell, with the increase of slip, to 62.4 per cent. at 25 knots and 59.4 per cent. at 26 knots. This is a remarkable result for a turbine equipment, and comes pretty near to the efficiency of the crack German liners, which have shown as high as 67 and 68 per cent. The present propellers were adopted after a series of trial runs with four different designs of propellers: one by the Navy Department; another by the Denny firm, Scotland; a third by the Vulcan Works, Germany; and the fourth by the Fore River Company. The Government design broke down through excessive cavitation early in the trials. The Denny propellers showed 50 per cent. efficiency at 24 knots, the Vulcan 54.04 per cent. at 24 knots, and the Fore River type, which was designed by the chief engineer, Mr Charles B. Edwards, showed 62.7 per cent. at 24.5 knots. We present two illustrations of these propellers, which are 9 ft. 6 in. diameter, with a pitch of 8 ft. 8 in., that will possess strong interest in connection with these comparative figures.

"The standardisation trials held to determine the number of revolutions of the propellers corresponding to various speeds, from 12 knots to the highest speeds of the vessel, took place off Rockland in from 40 to 60 fathoms of water. The start and end of the mile are marked by pairs of posts set up on shore, and the time is taken from the



Lowering Rotor into Casing of 12,000 H.P. Curtis Turbine.

Constructed by Fore River Shipbuilding Co., Mass., U.S.A.

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bridge from the moment that the first pair come in line to the instant that the finish line is crossed. Meanwhile the revolutions of the engines are accurately recorded by a mechanical counter. The effect of the tide, whose velocity is measured by a Government vessel stationed at the centre of the course, is eliminated by making the alternate runs with and against the tide. The "Salem" made five runs over the course, the fastest, with a favourable tide of 0.8 of a knot, showing a speed of 26.88 knots an hour, and the mean of all five runs working out at 25.957 knots. The mean displacement during the runs was 3,745 tons. On the fastest run of 26.88 knots, the propellers made 382.4 revolutions per minute. The steam pressure at the steam chest on the turbines was 253 lbs. The peripheral speed of the blades, at the above speed, was 12,000 ft. per minute, and the horse-power was 20,200, or over 25 per cent. more than was required by contract. It was estimated that the ship would make 24 knots with sixteen nozzles open on the turbines; but she actually made 25.4 knots under these conditions, and 26.88 with the full number, twenty, open. The coal used on these trials was a screened Pocahontas.

"In the starting and stopping trials the engines went from full speed ahead to full speed astern in 1 minute and 30 seconds, and from full speed astern (at which they develop 70 per cent. of the full speed ahead power) to full speed ahead in 1 minute and 4 seconds."

. . . .

"Due to the strong blade construction and large blade clearances, the Curtis turbine can operate without damage when large quantities of water are carried over from the boilers. It can thus operate under conditions which would wreck a reciprocating engine, the only effect on the turbine being a reduction in its speed.

"Due to the above-mentioned ability to stand water, and to the small effect of heat expansion, the Curtis turbine can be started at once from a cold condition in case of necessity, while the reciprocating engine requires a gradual warming up.

"The following are the chief advantages claimed for the Curtis turbine over that of the Parsons type:—

"(1) Due to the use of the impulse principle and the compounding by velocity stages in addition to compounding by pressure stages, the number of blades required to obtain a given economy is very much reduced, being only about one-fortieth as many.

"(2) The blades are made of a very much heavier section and are held in the wheels and casing in a more solid and substantial manner. This, in addition to the protection afforded by the design of the shrouding and supporting base, makes the construction so substantial that the rotating part may be moved forward or aft until actual contact occurs between the moving and stationary parts without any damage being done. Similar treatment on the Parsons design would result in stripping the blading.

"(3) The use of the impulse principle obviates the necessity of

small clearance at the ends of the blades, as there is no tendency for the steam to leak around the ends. A large clearance can thus be used, greatly improving the mechanical reliability.

"(4) With the Curtis design one turbine only on each shaft is necessary for economical operation at all speeds; and equal results can be obtained as with a Parsons installation using two extra cruising turbines, one high and one intermediate pressure.

"(5) Full steam pressure only comes on the steam chest, which is a comparatively small steel casting, and the greatest pressure in the turbine shell is less than one-third the full working steam pressure. This permits the use of a high working pressure and large turbine diameters, and greatly reduces the chance of any difficulty from heat expansion. It also permits the turbine to be started very quickly from the cold condition without danger of unequal expansions causing trouble, while the Parsons turbine requires gradual warming up and careful watching of the clearance during the process.

"(6) Absence of dummy pistons and their packing eliminates the leakage of high-pressure steam and makes the economy independent of any adjustments, so that the initial economy will be maintained continuously. It also eliminates the necessity of maintaining the very fine axial adjustment required for dummy-piston packing strips, and the danger of their fouling.

"(7) Under given conditions the Curtis will run at lower revolutions, thus permitting twin screws to be used. This gives a higher propulsive efficiency and makes the entire propeller area available for backing. It also greatly simplifies the entire machinery installation; especially controlling valve arrangement, which gives much better manœuvring qualities.

"(8) Due to the simple controlling-valve arrangement used with twin screws, the large clearances around blades, the absence of dummy-piston packing, the substantial blade construction, and the low steam pressure in shell, as above described, it is possible to throw the turbine instantly from full ahead to full astern with entire safety. In the Parsons design reversing must be done with considerable care to avoid damage.

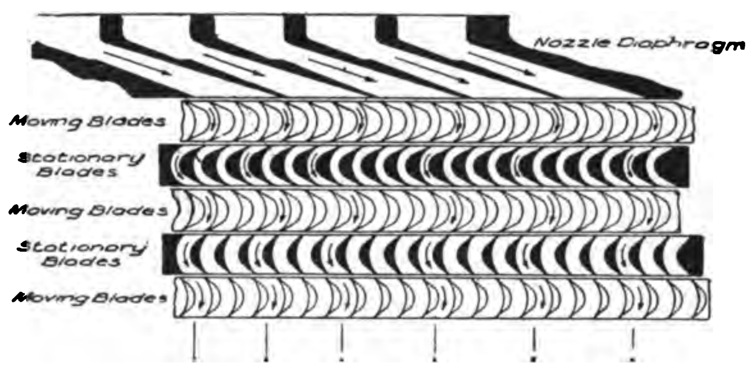
"(9) As previously described the Curtis turbine can obtain speed control by using either the throttle or steam-chest nozzle valves, or a combination of both, while the Parsons turbine only has the throttle. Also in twin-screw installations, as used by Curtis, each turbine is independently controlled with great certainty, while in Parsons installations, with one turbine exhausting into another, each screw cannot be independently controlled, and actual speed of the vessel cannot be so accurately known.

"(10) As above described the Curtis turbine can handle without damage large doses of water in the steam supply, which would strip the blading of the Parsons design."

. . . .

Principle of Operation.—"The Curtis turbine is of the impulse type, and, in order to reduce the economical speed of rotation to a point suitable for direct driving of marine propellers, the turbine is divided into several pressure stages, and each pressure stage is provided with several rows of revolving buckets. The simplest possible impulse turbine would consist of a single wheel having a single row of buckets on which a jet of steam is directed by a suitable nozzle. Examples of such a turbine are the De-Laval steam turbine and the Pelton water wheel.

"A simple single-wheel impulse turbine has its maximum economy when the buckets move with a velocity of approximately half that of the jet of steam. A jet of high-pressure steam, in a proper nozzle, will attain a velocity of about 4,000 ft. per second when discharging into a vacuum, which would require the enormous speed of 2,000 ft. per second for the buckets to obtain the maximum economy. This speed would require such a high number of revolutions that it would



"Stage" Diagram of Curtis Type Turbine.

render direct connection to a propeller impossible, and would also produce centrifugal forces too great to be properly handled.

"By using several wheels on the same shaft, each one utilising the exhaust from the one preceding, the pressure drop and corresponding steam-jet velocity to be handled by each wheel is greatly lessened. Thus by using eight wheels, each in a separate chamber with its own nozzle, and so proportioning the steam pressure that the available energy is equally divided, the steam-jet velocity is reduced to about 1,400 ft. per second and the bucket speed to 700 ft. per second. This, however, is still too high for satisfactory marine work, and, in order to lower it, each wheel is provided with three rows of moving buckets instead of only one. Each row of moving buckets will take out from the jet velocity approximately twice its own speed, so that the use of three rows on each of eight wheels will give a bucket speed of 230 ft. per second. This speed can be used satisfactorily on high-speed vessels, but as the economy curve is quite flat at the point of

maximum efficiency, the speed can be somewhat reduced without greatly reducing the economy, giving a fair working speed of about 160 ft. per second. The above-described method of reducing the economical bucket speed by means of several separate stages, each having several rows of buckets, is illustrated in diagrammatic form which shows the first two stages of a turbine of which the remaining stages would be similar. It is seen that between each row of moving blades there is a row of stationary blades, the object of which is to reverse the direction of flow of the steam as it leaves one row of moving blades and direct it into the next row of moving blades. The arrows show the direction of the steam as it passes through the nozzle and then through the moving and stationary blades.

"When the steam passes through a nozzle its pressure drops and it gains velocity; this velocity is absorbed by the moving buckets, thus converting energy of the steam into mechanical work. As the steam passes from stage to stage it drops in pressure, and when it passes from one row of buckets to the next in the same stage it loses velocity, but the pressure is the same in any one stage in all the bucket rows, and is also the same on both sides of the wheel.

"This uniform pressure in all parts of stage obviates the necessity of small radial clearance between the ends of the blades and the casing, since there is no leakage around them. This clearance is, therefore, made as large as desired for proper mechanical construction.

"The axial clearance, or distance between the edges of the blades, is also made quite large, its only limit being the slight disturbance of the jet which occurs at some distance from a nozzle or guide blade."

Description of Construction of "Salem's" Turbines.—The plate opposite is a vertical section of one of the turbines as fitted to the "Salem," and clearly shows the construction. The turbine consists of a cast-iron cylindrical casing divided by dished cast-iron diaphragms into a series of separate compartments (cast steel is used for two first stages, for strength). In each compartment or "stage" there is a separate wheel which carries on its periphery three rows of moving buckets (for reasons later described the first wheel has four rows). The wheels are all mounted on a hollow steel shaft carried by self-aligning bearings as shown. Where the shaft passes through the diaphragms they are provided with bronze bushings having a small clearance, thus preventing appreciable steam leak from one stage to the other. Where the shaft passes out through the ends of the casing it is provided with carbon stuffing boxes, which prevent steam leaking out at the ahead end or air leaking in at the back end where a vacuum exists.

The stuffing boxes are supplied with steam in the space between the carbon packing to prevent air leaking in and lowering the vacuum. They are also drained to the fourth-stage shell.

Cast-steel steam chests for ahead and astern running are attached to the front and back casing heads as shown, and are flanged for the main steam pipes. The nozzles for each stage are bolted to the diaphragms as shown, the diaphragms having steam-port openings cast in them to allow the steam to pass through to the nozzles.

Manœuvring is accomplished by means of two lever-operated balanced throttle valves, each taking steam from the main steam pipe, and one delivering to the ahead steam chest and the other to the astern steam chest. There are seven ahead wheels and two reverse wheels. The reverse wheels are mounted in the after end of the casing, and under ordinary ahead running they are in a vacuum and therefore do not waste power by steam friction. They are the same as the ahead wheels, except the blades are reversed. To reverse

A B C D

Blading Segment of Curtis Turbine.

A, Blades held in core previous to casting on Base and Shroud.
B, Blade cut to length and notched.
C, Segment after casting on base and shroud.
D, Segment machined to size.

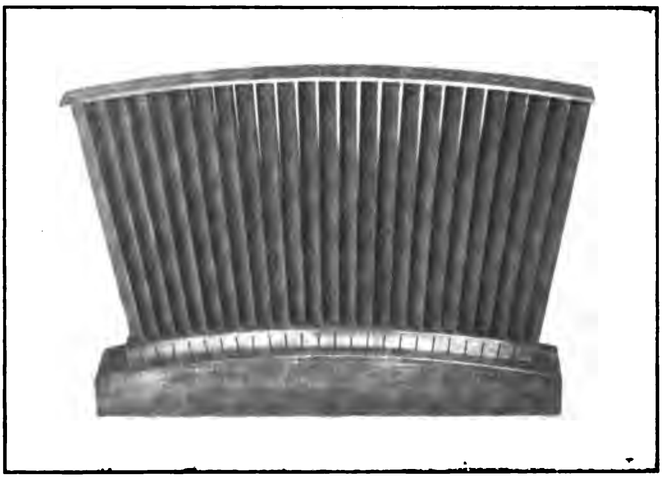
when going ahead the ahead throttle valve is shut and the reverse throttle valve opened, which is easily and quickly accomplished by the operating levers of the two throttle valves.

In the ahead steam chest there are twenty valves, each opening one of the nozzles for the first-stage wheel. For continuous running sufficient of the nozzles are opened to give the speed desired, and the ahead throttle valve is left open, thus giving full pressure in the steam chest. Sixteen nozzles will give full designed power, leaving four for overload. The astern steam chest has the same number of nozzles, but only eight have valves. For manœuvring, the nozzle valves in the steam chests are left open and the speed is controlled by the throttle valves.

Drain pipes are provided connecting each stage with the next, so the condensed steam in any stage will pass to the next one of lower

pressure and there give up a part of its heat to do useful work. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. A regular marine thrust bearing is attached to the forward end of the turbine shaft. In addition to taking the propeller thrust this bearing also maintains the proper axial position of the rotor, so that the axial clearance of the blades is correct. This clearance is $\frac{1}{10}$ of an inch on the first wheel and increases to $\frac{1}{4}$ of an inch on the seventh wheel. This axial clearance is very ample to allow for all unequal heat expansion that may occur and any mechanical irregularities, and leave sufficient leeway for adjustment.

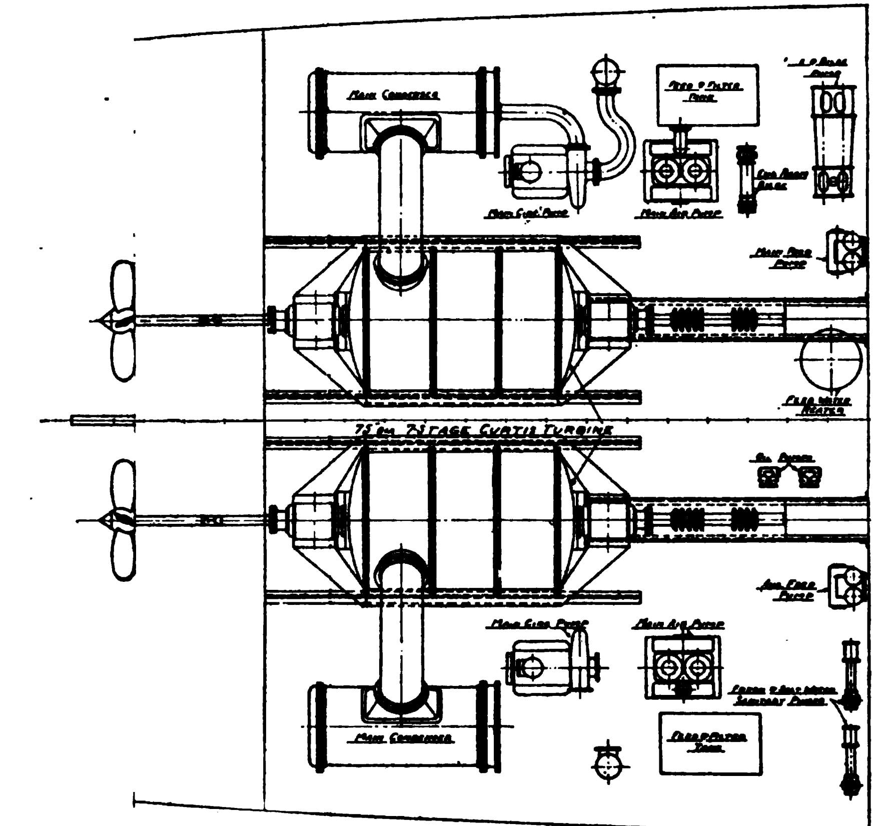
The construction of the blading is shown in the illustrations. The blades are made of special bronze accurately formed to the required



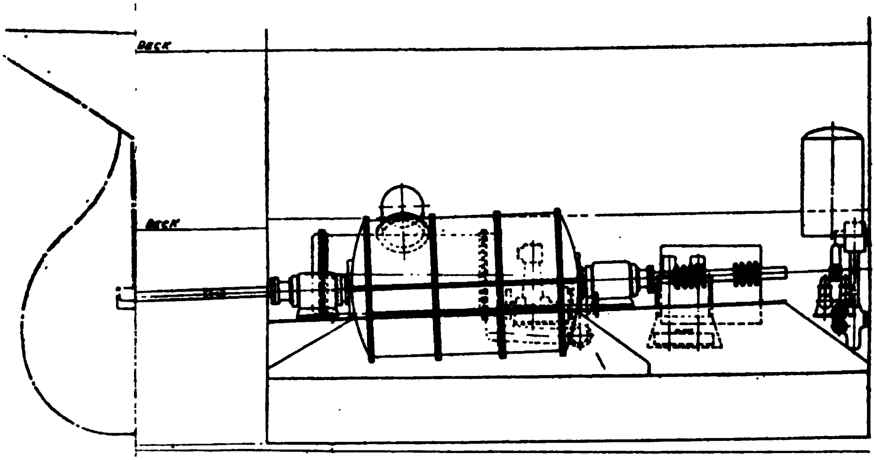
Latest Type Curtis Turbine Blading.
Consisting of Steel Channel Base with saw cuts to allow of flexibility in fitting, Steel Plate Shroud, and extruded Metal Blades.

shape by being extruded through a die and thus made into long bars. These are cut into lengths as required by each stage, and each length is notched or drilled through at the ends, as shown at B. A number of blades are built up into a segment by casting on a composition base on the inner ends and a shroud on the outer ends as shown at C. In order to do this casting the blades are held in a sand core, with the ends projecting slightly, as shown at A. After casting, the base and shroud are milled off true, thus completing the segment, as shown at D.

The ends of the blades fuse into the cast parts, thus making practically one solid piece of the entire segment. If any blade should not happen to fuse, the metal will flow into the notch or hole and effectually keep it in place. The extruding process gives the blades a hard, smooth skin, which reduces steam friction and effectually prevents wear and erosion of the blades.



PLAN



ELEVATION

[To face page 248.

The segments are held in the wheel rims by inserting the bases in rectangular grooves around the steel rims and caulking the edges of the grooves. This caulking is done by means of a pneumatic caulking tool mounted on special adjustable pedestal, while the wheel is slowly revolved.

The bases and shrouds are overhung slightly beyond the edges of the buckets, so that if by any accident the wheels should be moved axially the blades would not come in contact and be damaged, as the overhanging bases and shrouds would protect them. These bases and shrouds are very strong, and can stand forcing together so hard that the turbine would be brought to a stop without any serious damage occurring. This very important feature renders stripping of the blades a practical impossibility in this design of turbine.

As before stated, the radial clearance at the ends of the blades can be made as large as desired without having any effect on the operation, and, as constructed, it varies from $\frac{1}{4}$ in. to 2 in., depending upon the proximity of the stationary blade holders, as is clearly shown in the illustration.

To allow for the increased volume of the steam as it expands in passing from stage to stage at lowering pressures, the length of the blades is increased, as shown in sectional view, and also the arc of the nozzles is increased, thus giving greater area of passage in each succeeding stage. Also, in any *one* stage, the blade lengths are increased in each succeeding row, because the velocity falls as the steam passes from row to row, although it is at practically constant pressure throughout the stage.

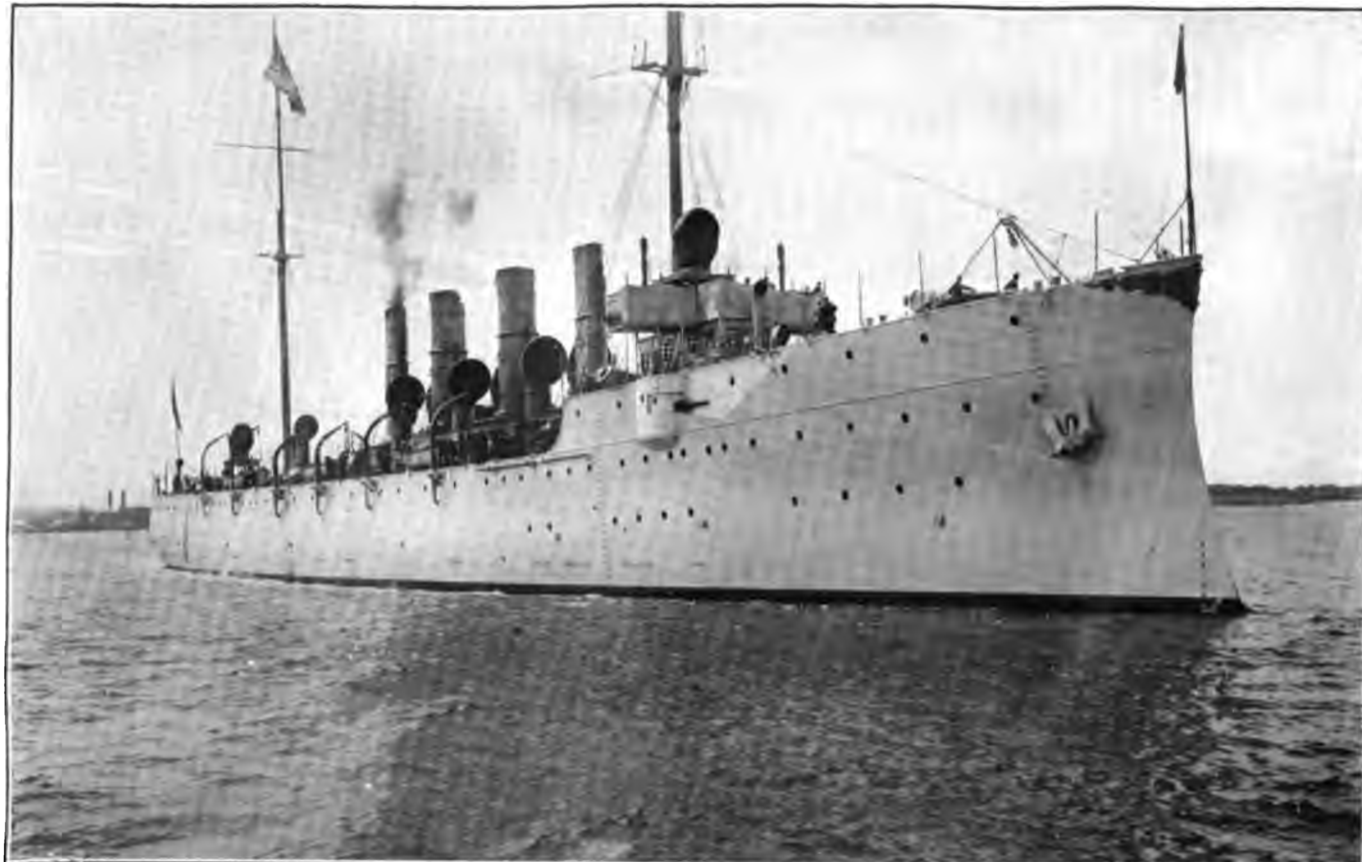
In order to keep the pressure in the shell as low as possible, the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the first stage, and one-eighth in each of the other stages. This requires the first-stage nozzle to be of the expanding type, but all the other nozzles are of the parallel-flow type. Also the first-stage wheel is provided with four rows of buckets instead of three as on the other wheels, since the greater energy drop produces greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. This arrangement makes all the ahead wheels except the first operate under eight-stage conditions.

Reciprocating and Turbine Engines compared.—The results shown below of the first series of competitive trials between three American vessels of the "cruiser scout" class afford an interesting comparison of performances, and prove conclusively the superiority almost throughout of the turbine over the ordinary reciprocating engine.

The "Birmingham" is fitted with reciprocating engines, the "Salem" with Curtis type turbines, and the "Chester" with Parsons type turbines, otherwise the three steamers are identical in dimensions and displacement. The value of the results is greatly discounted by the fact that the shaft horse-power of the "Chester" could not be

Cruiser Scout Trials.		"Birmingham."	"Salem."	"Chester."
Standardisation	Date	March 11	June 23	Feb. 28
Best uncorrected run, knots		25.192	26.886	26.22
Mean of best pair of runs, knots		24.477	26.11	25.138
Mean of five high runs, knots		24.236	25.957	25.074
R.P.M. required for 24 knots		187.23	335.2	507.25
R.P.M. required for 22.5 knots		170.38	312	466.4
R.P.M. required for 12 knots		89.7	165	245.5
Full speed, four hours	Date	March 12	June 25	March 1
Mean displacement		3,720	...	3,673
Mean speed		24.325	25.947	26.52
Mean R.P.M.		191.66	378.39	614.31
Apparent mean slip, per cent		15.8	19.8	25.6
Coal per hour, lbs.		29,904	38,502	38,332
Indicated horse-power		15,540	21,333*	...
Admiralty coefficient		222.5	197.5	...
Brake horse-power		...	19,200	Unknown.†
Coal per I.H.P. per hour		1.92	1.81*	...
Coal per B.H.P. per hour		...	2.01	?
Miles per ton of coal		1.822	1.51	1.548
Radius (nautical miles)‡		1,730	1,434	1,471
Twenty-four hours at 22.5 knots	Date	March 14	June 30	March 3-4
Mean displacement		3,741	...	3,716
Mean speed		22.665	22.536	22.779
Mean R.P.M.		172.1	312.535	473.34
Apparent mean slip, per cent.		12.6	15.7	18.7
Coal per hour, lbs.		20,510	18,485	18,063
Indicated horse-power		10,760	10,378*	...
Admiralty coefficient		260 9	266.5	...
Brake horse-power		...	9,340	Unknown.†
Coal per I.H.P. per hour		1.91	1.78*	...
Coal per B.H.P. per hour		...	1.97	?
Miles per ton of coal		2.475	2.73	2.83
Radius (nautical miles)‡		2,348	2,593	2,686
Twenty-four hours at 12 knots	Date	March 13	June 26-27	Feb. 28-29
Mean displacement		3,743	...	3,710
Mean speed		12.228	11.937	12.2
Mean R.P.M.		91.4	164.11	249.7
Apparent mean slip, per cent.		11.2	15.15	17.5
Coal per hour, lbs.		4,629	4,051	4,091
Indicated horse-power		1,600	1,511*	...
Admiralty coefficient		275.2	271.7	...
Brake horse-power		...	1,360	Unknown.†
Coal per I.H.P. per hour		2.89	2.68*	...
Coal per B.H.P. per hour		...	2.98	?
Miles per ton of coal		5.965	6.6	6.66
Radius (nautical miles)‡		5,643	6,269	6,338

* Equivalent I.H.P. based on assumption of 10 per cent. engine friction.
† Torsion meter gave totally unreliable readings, going as high as 28,000 horse-power at full speed.
‡ Starting with 950 tons of coal.



United States Scout Cruiser "Salem." Twin Screws, Curtis Marine Reversible Turbines, 20,000 H.P. Maximum.

[*To face page* 250.

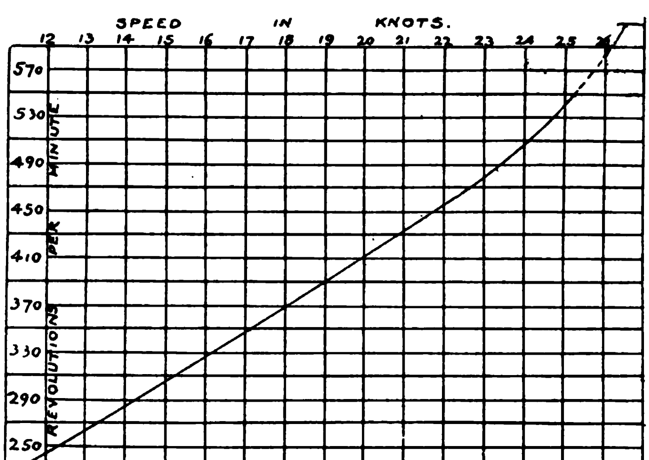
"Chester."

(Parsons Turbines.)

Twenty-four Hour Endurance Trial at 12 knots.

Average Pressure (gauge).	Forward Engine-Room.	After Engine-Room.
Main steam - - -	15.5 lbs.	147.4 lbs.
H.P. turbine - -	10.3 „	8.2 „
L.P. „ - -	23.41 in. vac.	23.4 in. vac.
H.P. cruising - -	83.2 lbs.	...
I.P. „ - -	20.9 „	...
Condenser - - -	29 in. vac.	28.8 in. vac.

Barometer - - - - - - - - - - - 29.75 in.
Four boilers used, grate surface - - - - - - 232 sq. ft.
Coal per sq. ft. of grate per hour (natural draft) - - 17.7 lbs.



Revolutions and Knots Curve.

U.S. Turbine Cruiser Scout "Chester."

Twenty-four Hour Endurance Trial at 22.5 knots.

Average Pressure (gauge).	Forward Engine-Room.	After Engine-Room.
Main steam - - -	230 lbs.	230 lbs.
H.P. turbines - -	98 "	93 "
L.P. " - -	0 "	2.25 "
H.P. cruising - -	19 in. vac.	...
I.P. " - -	180 lbs	...
2nd H.P. "expansion" -	53 "	50 lbs.
Condenser - - -	29.9 in. vac.	29.3 in. vac.
Auxiliary exhaust - -	0	2 lbs.

Feed temperature - - - - - - -	185 degs. Fahr.
Injection - - - - - - - -	35 "
Discharge - - - - - - - -	64 "
Coal per sq. ft. of grate per hour - - - -	26 lbs.
Air pressure - - - - - - - -	.7 in.

Four Hour Full Speed Trial.

Average Pressure (gauge).	Forward Engine-Room.	After Engine-Room.
Main steam - - -	243.5 lbs.	246.7 lbs.
H.P. turbine - -	238.7 "	239.5 "
2nd H.P. "expansion" -	159.5 "	149.8 "
L.P. turbine - - -	15.5 "	15 "
H.P. cruising - -	20 in. vac.	...
I.P. " - -	20 "	...
Astern turbine - -	28 "	28 in. vac.
Condenser - - -	28.8 "	28.3 "

Port Propeller of U.S. Curtis Turbine Scout Cruiser "Salem."

Pitch, 8 ft. 8 in. Diameter, 9 ft. 6 in. Speed, 26.8 knots. Propulsive efficiency at 26 knots, 59.4 per cent. (referred to B.H.P.).

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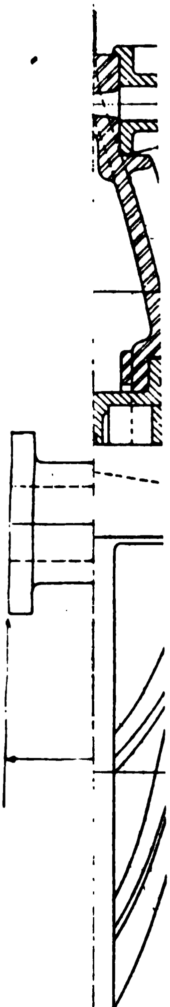

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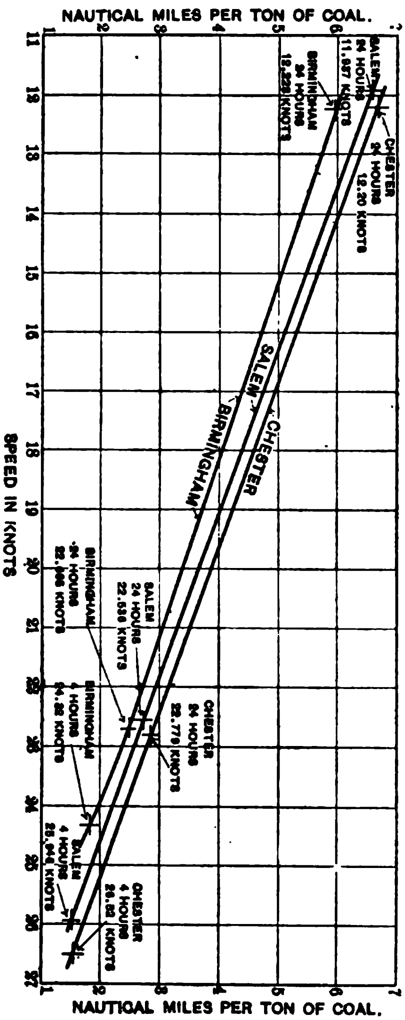
measured, owing to breakdown of two of the torsion meters, and the comparison of power and consumption at the progressive speeds is therefore unknown. Otherwise the results are of particular interest, and it will be noticed that the "Chester" takes leading place, both



CURVES FOR SCOUT CRUISERS, SHOWING NAUTICAL MILES PER TON OF COAL AT VARIOUS SPEEDS.

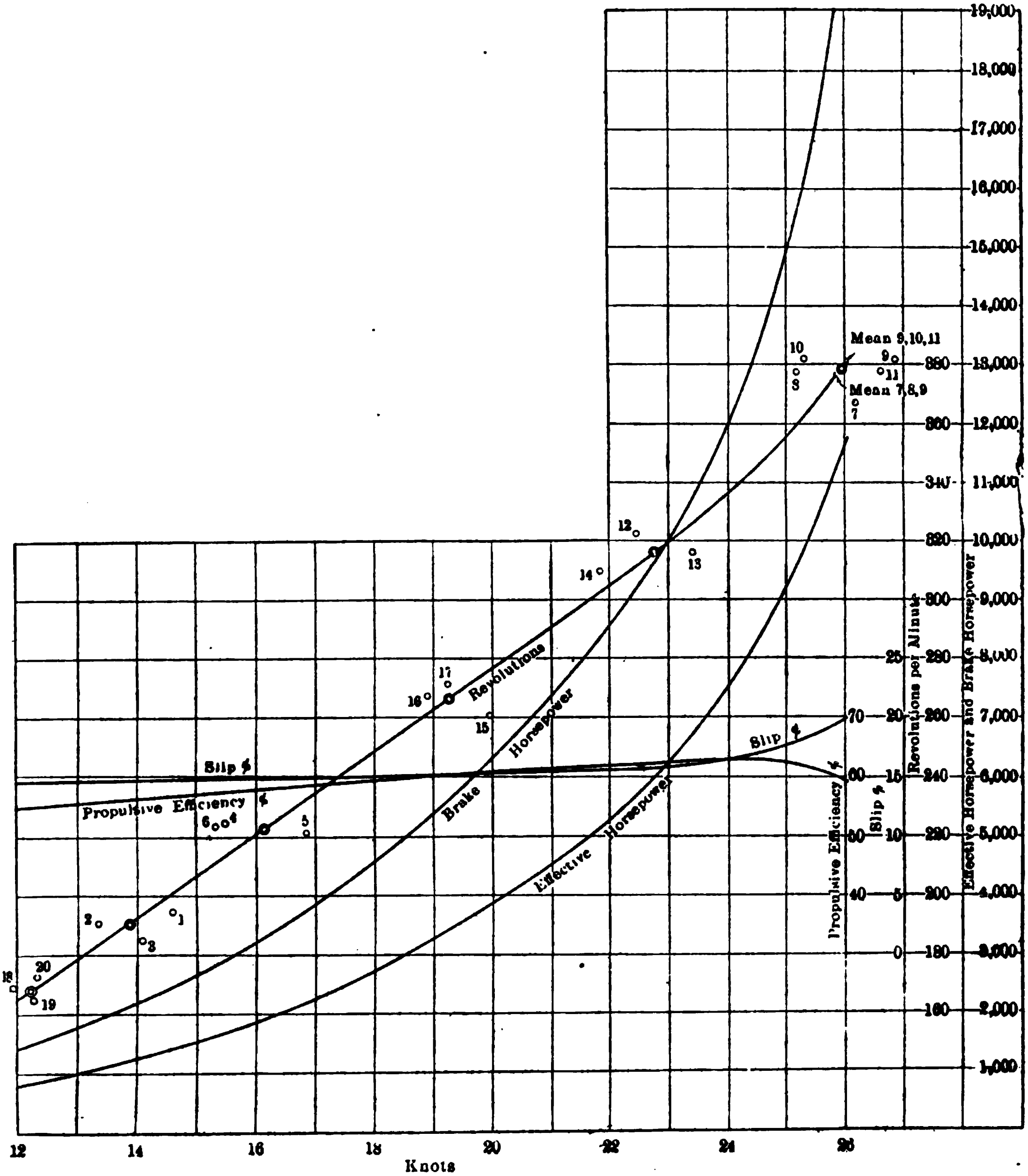
as regards consumption and speed, with the "Salem" a good second. At the same time it might be said that the reciprocating engine vessel does not appear so economical as usually found in good practice, as 2.89 lbs. of coal per I.H.P. hour at 12 knots is certainly

abnormal, even allowing that the engines were designed for the higher speed of 24 knots, as with reciprocating engines there should be very little difference in consumption per I.H.P. at either high or low powers and speeds.

On referring to the trials of the U.S. cruiser scout "Salem," it will be noticed that four sets of propellers were tried before these, giving the best results, were discovered. From this it may be assumed that propeller design is still in its infancy. It is, however, encouraging to know that propellers of suitable and efficient design *can* be produced to meet the severe conditions imposed by high turbine revolution speeds, as, at maximum speed, the "Salem" turbines were running at nearly 400 revolutions per minute, and the propellers giving out a propulsive efficiency of 62.7 per cent. (referred to the shaft horse-power)—a figure abnormally high for turbine propellers, and perhaps never previously attained except in the case of reciprocating engines. Future improvement in propeller design will result in higher combined efficiency of turbines and propellers with a correspondingly reduced coal and steam consumption.

It is now common knowledge that a prominent Clyde firm is at present engaged in experimenting on turbines of the Curtis type, with the ultimate object of fitting them in a Government vessel at present under construction; the experiments will doubtless be directed towards the reduction in weight, which at present forms a rather serious objection to the Curtis type of turbine, and to improvements in consumption, as the economy is not what might be desired. The superiority of the Parsons turbine over the Curtis type lies chiefly in the reduced weight (turbine for turbine of a given power), and in reduced steam and coal consumption per horse-power hour, this latter being, of course, the crucial test from a commercial point of view. In the standard Parsons arrangement the total expansion of the steam is carried out in three turbines, one H.P. and two L.P., the H.P. terminal pressure (and the L.P. initial pressure) being usually from 12 to 20 lbs. gauge, or from 27 to 35 lbs. absolute pressure; whereas in the Curtis turbine the total expansion is effected in two single turbines of equal size, each receiving direct boiler steam independently and exhausting right down to the condenser pressure and temperature. Again, in the Parsons turbine, the H.P. casing is subjected to the full boiler pressure (less "drop"), while in the Curtis turbine the boiler steam expands and drops in pressure in the admission nozzles, so that the pressure on the casing is much less (see Table I., page 233). This expansion and pressure drop occurs at the initial end of each stage by means of the fixed nozzles through which the steam leaves one stage chamber to enter the next. It should also be noted that the steam in passing the guide blades of a Parsons turbine does work on itself to increase its own velocity, whereas in the Curtis type no such work requires to be done, the blade angles and openings being merely designed to guide or direct the flow of the steam from ring to ring of moving blades. This naturally results in a falling off in the velocity of the steam through the fixed and moving blades, and the required stimulus to the speed of the steam is obtained in the expanding nozzles of the succeeding stages, and this condition maintains throughout the turbine. An examination of the stage diagram (page 245) will make clear the differences in the blade angles and openings as compared with the Parsons system (page 15). Perhaps the chief reason for the excessive weight of the Curtis turbine lies in the necessity for having to continue the shaft or spindle right through the casing from end to end instead of finishing it off at the "wheels" as in the Parsons type. The blading system is, however, also to the heavy side, and the separate wheels to which the moving blades are fixed are necessarily somewhat massive in construction.

In the Curtis turbine the loss by blade tip leakage is very small as the leaking steam in each stage passes through the inlet nozzles of the next stage and is therefore utilised later on, the only actual loss being that of the last stage, where, however, the pressure is at a minimum. Leakage losses will occur, however, at the diaphragms or divisions between the successive stages where the revolving shaft passes through, although this has been reduced by screwing the holes in the diaphragms, which device, it will be noted, is similar in effect to the "fin" arrangement of the Parsons type gland rings.



Standardisation Curve.

U.S. Turbine Cruiser Scout "Salem."

Note.—Observe that at a speed of 23 knots the brake horse power curve is cut by the vertical ordinate at 10,000, and the effective horse power curve cut at about 6,250 horse power, then, 6,250 ÷ 10,000 = 62.5 per cent. propulsive efficiency at this speed.

SECTION IV.

TORSION METERS, &c.

The Bevis-Gibson Flash-light Torsion Meter.

THE following extracts from a paper, entitled "Torsion Meters as Applied to the Measurement of the Horse-Power of Marine Steam Turbines," by J. Hamilton Gibson, Esq., and read on 24th January 1908 before the North-East Coast Institution of Engineers and Shipbuilders, describe the chief features of this flash-light type of torsion meter:—

"When a revolving shaft transmits power it always twists slightly throughout its length. In other words, the end at which the power is applied moves slightly in advance of the end where the work is done, the amount of twist varying directly as its length, directly as the moment of the load applied, inversely as the rigidity of the material, and inversely as the fourth power of its diameter, the formula reading—

$$\theta = \frac{10.2\ TL}{CD^4},$$

where θ is the angular displacement in radians, T = twisting moment in inch-lbs., L = length of shaft in inches, C = the modulus of rigidity, and D = diameter of shaft in inches. The law holds good absolutely for all shafts which are not stressed beyond the elastic limit. As shafts are usually designed with a large factor of safety, it follows that the amount of twist, or the 'torque,' as we prefer to call it, is very small. In propeller shafting, for instance, the torque is rarely more than 1 deg. for 10 ft. of length, so that for a 12-in. shaft the circumferential displacement is only about $\frac{1}{8}$ in. at full power.

"Various methods and numerous instruments have been devised to enable an observer to read off the torque of revolving shafting, and such instruments are rightly termed 'torsion meters,' or, if self-registering, 'torsion indicators.'"

. . . .

"Before applying any form of torsion meter to a shaft, we must

know its 'modulus of rigidity'—that is, how much it will twist with a given static load applied at the end of a lever of known length. This can only be done satisfactorily in the workshop, preferably on a long rigid lathe-bed. One end of the shaft is securely fixed and a twisting moment applied at the other end. To eliminate the effect of friction in the supporting bearing at the free end it is advisable to use two levers, one at either side, as shown in the illustration, and the loads are preferably applied by graduated spring balances. Two pointers independent of the load levers are secured to the shaft, as far apart as practicable, and the difference in the angular movement of these two pointers gives the true twist for that length of shaft. If the pointers are made 57.3 in. long from the shaft axis, their ends will describe 1 in. of arc for 1 deg. of twist, and a decimally-divided straightedge will then measure the twist to within $\frac{1}{100}$ deg., which is quite near enough for all practical purposes, and we can proceed to calculate the modulus of rigidity from the formula.

"Observe that a propeller shaft is subject to two distinct stresses. Not only is it twisted as between the engine and the propeller, it is also compressed longitudinally by the propeller thrust, the compressive stress being sometimes as much as 20 per cent. of the shear stress at the surface of the shaft produced by torsion alone. This compression augments the torque by an appreciable amount, which has been actually measured in numerous experiments, and may be taken roughly as 3 per cent. for hollow shafts and 1 per cent. for shafts which are solid. It might be considered sufficient to calibrate only one shaft in a multiple-screw vessel; but it is found that similar shafts, with identical tensile and elongation tests, have different moduli of rigidity, probably due to their varying elastic limits and some slight difference of homogeneity in the material. The only way, therefore, to ensure accuracy is to calibrate each shaft separately and to build up a power diagram, as shown in Fig. 1, for each.

"Another point to bear in mind is that a working propeller shaft is 'alive,' and this condition must be imitated as far as possible during calibration, by jarring the shaft with repeated blows of a mallet, so as to keep the mass in a state of molecular vibration. Otherwise the phenomenon of mechanical hysteresis, so marked in some static experiments, will obtrude itself and vitiate the results. Having established the true modulus of rigidity for each shaft, we may proceed to build up our power diagram based on the formula—

$$H = \frac{\theta\, D^4 N}{CL},$$

where H = shaft horse-power, θ = torque in degrees, D = diameter of shaft in inches, N = number of revolutions per minute, C = constant varying with the modulus of rigidity, and L = length of shafting in inches. In this formula we have all the elements for obtaining the shaft horse-power, and it only remains to ascertain the number of degrees of torque by means of a reliable and accurate torsion meter."

The Bevis-Gibson torsion meter is thus described:—

"Two blank discs are mounted on the shaft at a convenient distance apart. Each disc is pierced near its periphery by a small radial slot, and these two slots are in the same radial plane when no power is being transmitted and there is no twist on the shaft. Behind one disc is fixed a bright electric lamp masked, but having a slot cut in the mask directly opposite the slot in the disc. At every revolution of the shaft, therefore, a flash of light is projected along the shaft towards the other disc. Behind the other disc is fitted the torque-finder, an instrument fitted with an eye-piece and capable of slight circumferential adjustment. The end of the eye-piece next its disc is masked except for a slot similar and opposite to the slot in the disc. When the four slots are set in line, a flash of light is seen at the eye-piece every revolution, and if the shaft revolves quickly enough the light will appear to be continuous. This effect is apparent at anything over 100 revolutions per minute. At lower speeds the flash is seen to be intermittent, but this in nowise affects the accuracy and reliability of the result. At each end of the shaft, therefore, we have what is virtually an instantaneous shutter fixed, be it noted, directly to the shaft, and there is no connecting link or gear between the discs, either mechanical or electrical, except the beam of light which flashes once in each revolution clear through the two shutters. Let us suppose now the shaft to be transmitting power. One disc lags behind the other by a definite amount, and although three of the slots are still in line, the fourth slot, namely, that in the lagging disc, effectually blanks the flash and no light is seen at the eye-piece.

Flash-light Torsion Meter Results.

Engines, No. 1215.
Vessel, H.M.S. "Indispensable." Date, 31st November 1907.
Trial, official full power. At Clyde.

Shaft.		Run.	Steam at H.P. Receiver.	Revs. per Minute.	Reading.	Degrees Torque.	Shaft Horse-Power.
Starboard	Wing	IV.	180 lbs. sq. in.	710	5.90	1.26	8,300
	Centre			680	6.02	1.02	7,480
Port	Centre			687	4.64	1.21	7,520
	Wing			707	5.74	1.23	8,270
Mean revolutions				696	Total		31,570

"This is where the function of the 'torque-finder' comes in. To pick up the light again the eye-piece must be moved by an amount equal to the circumferential displacement of the lagging disc. This is accomplished by manipulating the micrometer spindle of the torque-finder, on which is a scale and vernier graduated in degrees. While the scale is fixed its vernier moves with the eye-piece, and the graduations are so marked that by the aid of a simple microscope conveniently hinged, differences of $\frac{1}{100}$ of a degree can be readily discerned. For shafts of ordinary size the scale is set at 13.6 in.

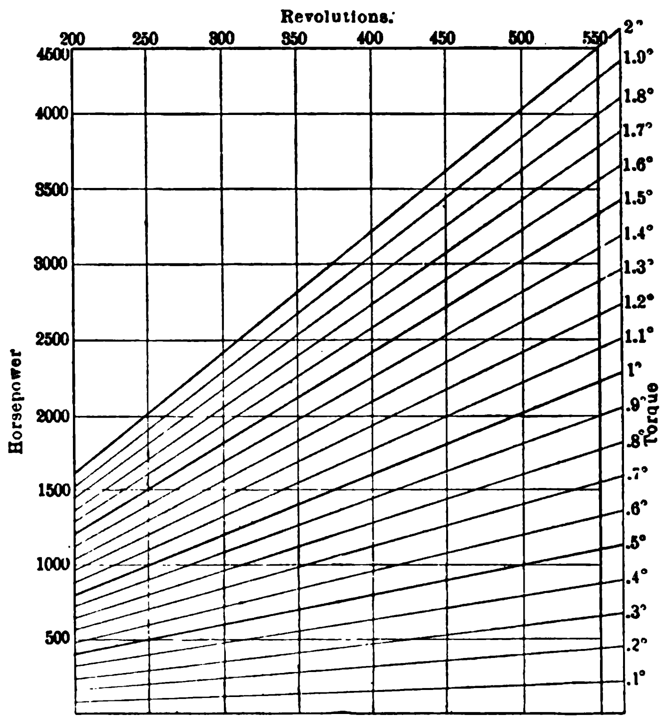
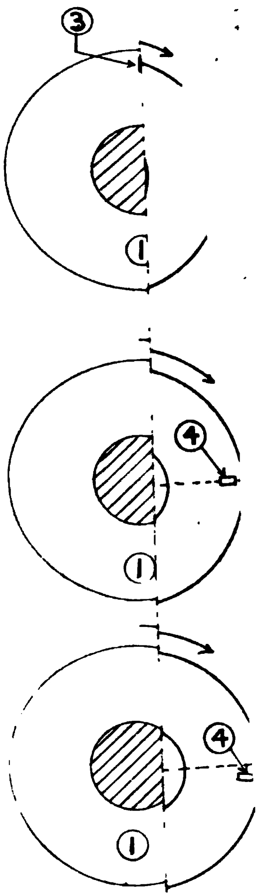


Fig 1.—Horse-Power and Torque Diagram.
(Constructed from Calibration Curve of Shaft.)

radius from the centre of the shaft, so that the degrees are about $\frac{1}{4}$ in. apart. One-hundredth of a degree, therefore, means $\frac{1}{100}$ of $\frac{1}{4}$ in., or $\frac{1}{400}$ of 1 in. As an ordinary shaft twists one degree in 10 ft. at full power, it is, therefore, possible to get the shaft horse-power to within 1 per cent. of full power. But as it is frequently possible to fit the discs 40 or 50 ft. apart, even this accuracy may be improved upon, and powers ascertained to within one-fourth of 1 per cent. of full power.

"The slots in the torsion-meter discs are necessarily of appreciable



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width, and in moving the torque-finder the light is visible for some distance along the scale. The light comes into view, attains a maximum amplitude and brightness, and fades away as the eye-piece

Turbine Shaft.	Degrees Torque.	Revs. per Minute.	Shaft Horse-Power.	S.H.P. Total.
Starboard L.P. - -	1.43	482.9	2,773	7,975
Centre H.P. - - -	1.69	461.2	2,600	
Port L.P. - - -	1.37	472.8	2,600	
Starboard L.P. - -	1.32	461.2	2,410	6,940
Centre H.P. - - -	1.65	426.8	2,330	
Port L.P. - - -	1.24	457.3	2,200	
Starboard L.P. - -	1.15	426.4	1,970	5,960
Centre H.P. - - -	1.52	417.6	2,080	
Port L.P. - - -	1.13	418.9	1,910	
Starboard L.P. - -	1.05	418.4	1,765	5,555
Centre H.P. - - -	1.52	422.3	2,120	
Port L.P. - - -	1.02	415.5	1,670	
Starboard L.P. - -	.21	198.6	162	495
Centre H.P. - - -	.27	206.3	185	
Port L.P. - - -	.19	183.5	148	
Starboard L.P. - -	.22	146.7	88	257
Centre H.P. - - -	.21	171.4	87	
Port L.P. - - -	.13	144.8	82	
Starboard L.P. - -	.07	46.3	13	37.2
Centre H.P. - - -	.05	86.1	15	
Port L.P. - - -	.01	24.4	9.2	

Fig. 2.—Actual Readings from a Flash-Light Torsion Meter and Corresponding Horse-Powers taken during Trial Trips of a Turbine Steamer under Varying Conditions of Displacement and Propellers.

moves along the scale. If it were possible to gauge the exact point where the light attains a maximum, that is the point that would be used. Failing this, however, use is made of one edge of the slot. The finder is moved always in the same direction in taking readings,

and stopped at the exact point where the light is cut off. So delicate is the sense of sight that a movement of $\frac{1}{100}$ of a degree is sufficient to mark the difference between light and darkness. A zero reading is taken when the shaft is revolving idly, if possible at or near full speed, and this reading forms a base and is subtracted from any subsequent power readings.

"First, suppose the shaft to be revolving idly. The finder is moved over until the light is just disappearing, and the vernier is seen to be standing at 0.53 degree. Now, suppose the shaft to be transmitting power. The discs have twisted relatively to one another, and no light is seen until the torque-finder is moved the same amount.

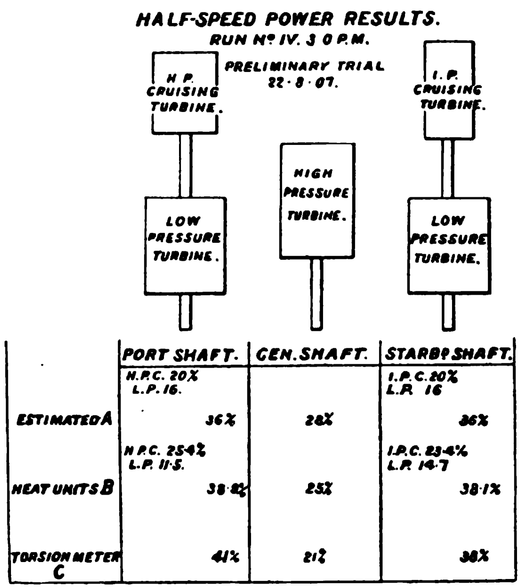


Fig. 3.—Percentage Distribution of Power for a Three-Shaft Turbine Installation.

Having picked up the light the finder is worked gently over until the light is just disappearing again. The reading is now 2.39 degrees. Subtract from this the zero reading of 0.53 degree and we get the true torque, namely, 1.86 degrees. Now, to apply our shaft horse-power chart, Fig. 1. We will suppose that the revolutions are 475 per minute. The torque is 1.86 degrees, and finding the intersection of the lines on the diagram the power is seen to be 3,620.

"It is perhaps scarcely necessary to point out that the whole operation takes much less time than its description. Indeed, it is possible to produce the shaft horse-power on a trial trip immediately on the termination of each measured mile run, and to hand a slip to

the officer in charge similar to that shown in Fig. 2, containing all the information, in plenty of time for him to make any necessary adjustments before coming back on the straight for the next mile.

"Then again the distribution of power in a turbine installation can only be approximately estimated. The steam is turned into the high-pressure turbine and left to follow its own devious course through the successive turbines on its way to the condenser. At low powers it is sometimes found that the high-pressure turbine shows the most power, while for over-loads the lower-pressure turbines have the advantage. In Fig. 3 the percentage distribution of power is shown by three sets of figures—for a three-shaft turbine installation, including high pressure and intermediate cruising turbines. Set A shows the estimated or designed distribution, set B the calculated distribution from the pressure-gauge readings at the terminals of each turbine, and set C shows the actual distribution of power over the three shafts as ascertained by flash-light torsion meter readings.

"Referring again to Fig. 2, it will be seen that the starboard low-pressure turbine shows throughout the series more power than the port. Investigation showed that the blade-tip clearances of the two turbines differed slightly, and a further comparison proved that the percentage difference of clearance was just sufficient to account for the difference of shaft horse-power recorded."

Turbine Advantages over Cylinder Engines.

Some of the advantages of the turbine over the ordinary reciprocating engine may be stated as follows:—

1. Fewer working parts, as no piston or slide valves, piston rods, &c. &c., are required.

2. Steam applied direct from the boiler to the shaft without any intervening loss of power.

3. Less danger of breakdown, as it is almost impossible for the small vanes to become damaged, unless actual contact takes place between the blades and casing, or between the rings of rotor and casing blades. In addition to this, three separate lines of shafting are provided, so that if one became disabled, two are still left for running with.

4. Less weight of machinery for the same power.

5. Engines placed well down in the vessel.

6. Greater speed for the same consumption of coal at high speeds.

7. Machinery vibration practically eliminated.

It may, however, be stated that the turbine is most economical in running at high speeds, as the difference in consumption between, say, 20 knots and 10 knots is not proportional to the difference in speed.

This is due to the fact that the most economical speed of the turbine bears a certain ratio to the velocity of the steam, in addition to the fact that the loss by tip clearance or leakage is more proportionally at low speeds.

With regard to the floor space occupied, there is not much to choose between the reciprocating engine and the turbine. In space taken up in the vertical direction, the turbine has a distinct advantage.

Regarding reliability, there is no reason why the turbine should be inferior to the reciprocating engine; and as to economy and speed the turbine has shown itself in many ways superior, although at very low speeds the reciprocating engine is superior in economy of steam; against this, however, the turbine requires less oil. The turbine, in its present state of development, is not suited for all classes of ships. From the weight point of view the turbine is superior to the reciprocating engine of the same power, particularly in the case of high-speed vessels, and possesses also the special advantage of being well balanced.

It may be some time yet before sufficient modification of the turbine allows of the adoption of this type of engine for the slow speed tramp steamer, but it is very probable that such modification will come, in perhaps the near future.

Denny & Johnson's Patent Torsion Meter.

The following description of the appliance is reprinted from *Engineering*, 7th April 1905:—

The diagram, Fig. 1, shows the general arrangement of the apparatus as applied to turbine-driven shafting. On the shaft, the torsion of which it is desired to measure, are fixed, at a suitable distance apart, two light gun-metal wheels A and B. On each wheel is mounted, as shown, a permanent magnet, the projecting pole of which is made V shaped, in order to produce a dense and definite magnetic field at the point. Underneath the magnets and set concentrically with the wheels and shaft are fixed two inductors A and B, each of which consists of a quadrant-shaped piece of soft iron carried on a gun-metal stand provided with suitable levelling-screws. On each piece of iron are mounted a number of separate but similar windings of insulated wire, there being a certain suitable number of windings per unit of circumferential length of the iron. There is in conjunction with the inductors a recording box, in which are mounted two series, A and B, of contact studs, around which are fixed scales. In connection with the series of studs two contact arms A and B are arranged, by means of which electrical connection may be made at will between any desired stud of a series and its contact arm. There is in series A a stud for every separate winding in the inductor A, and in series B a stud for every separate winding in the inductor B, each stud being connected to its particular winding by means of a separate wire, all the wires being contained in the multiple cables A and B. The remaining ends or returns of the winding on the inductors A and B are all connected by means of two common wires (also contained in the cables A and B) to the contact arms A and B respectively.

Included in each of these two circuits is a variable resistance (by means of which the strength of the current flowing in the circuit may be adjusted as desired) and one winding of a differentially-wound telephone receiver. The scale A is divided into six equal parts, there being six separate windings in the inductor A, and thus six studs in the series A; the length of five subdivisions of the scale thus represents the circumferential length occupied

by all the windings on the inductor, each subdivision representing the distance between the neighbouring windings, which is usually 0.2 in. The scale B is divided into fourteen equal parts, there being fourteen separate windings on the inductor B, and thus fourteen studs in the series B. The length of thirteen subdivisions of the scale, as before, represents the circumferential length occupied by all the windings in the inductor, the distance between neighbouring windings being represented by one subdivision of the scale; the usual distance between neighbouring windings in inductor B is 0.02 in. The wheel in connection with the inductor at the turbine end of the shaft is set so that the magnet fixed to it is exactly above one or other of the two end windings in the inductor.

The correct end winding to set the magnet to is that one from which the

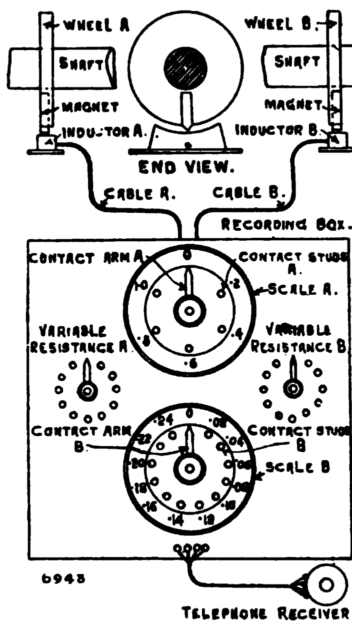


Fig. 1.

magnet will travel towards the other end winding when the shaft rotates. The wheel in connection with the other inductor is also set so that its magnet is exactly above one of the two end windings in the inductor, the correct end winding to set the magnet to in this case being that one from which the magnet will travel in the opposite direction to the other end winding when the shaft rotates. To facilitate the accurate and easy setting of the magnets above their respective windings, lines are cut in the tops of the inductors exactly above the end windings, and the magnets are set to these lines. When the shaft rotates without transmitting power, a current of electricity is induced in the end or zero winding of each inductor, the contact arms being first placed in contact with the end or zero stud in each series. These two separate currents both traverse their respective circuits, passing in each case from the inductor winding in which they are induced to the

respective zero studs to which these windings are connected, thence by way of the respective contact arms, resistances, and telephone receiver windings back to the inductors again. The connections to the receiver windings are so arranged that the effects of the two separate currents flowing therein are in opposition, and thus neutralise each other's effect on the receiver when the strengths of the two currents flowing are exactly equal at the same instant. By means of the variable resistances in each of the circuits the currents are made equal in strength, and then so long as the shaft transmits no power, and is thus subject to no torsion, no sound will be heard on listening at the receiver, since the currents induced in the zero windings of the inductors have been equalised, and are both induced at exactly the same instant. When transmitting power, the shaft is subject to a certain torsion or twist, which causes the zero winding of the inductor next the turbine or engine to be excited in advance of the other by the amount of the torsion of the shaft; a loud ticking sound will then be heard in the receiver, as the currents no longer neutralise each other.

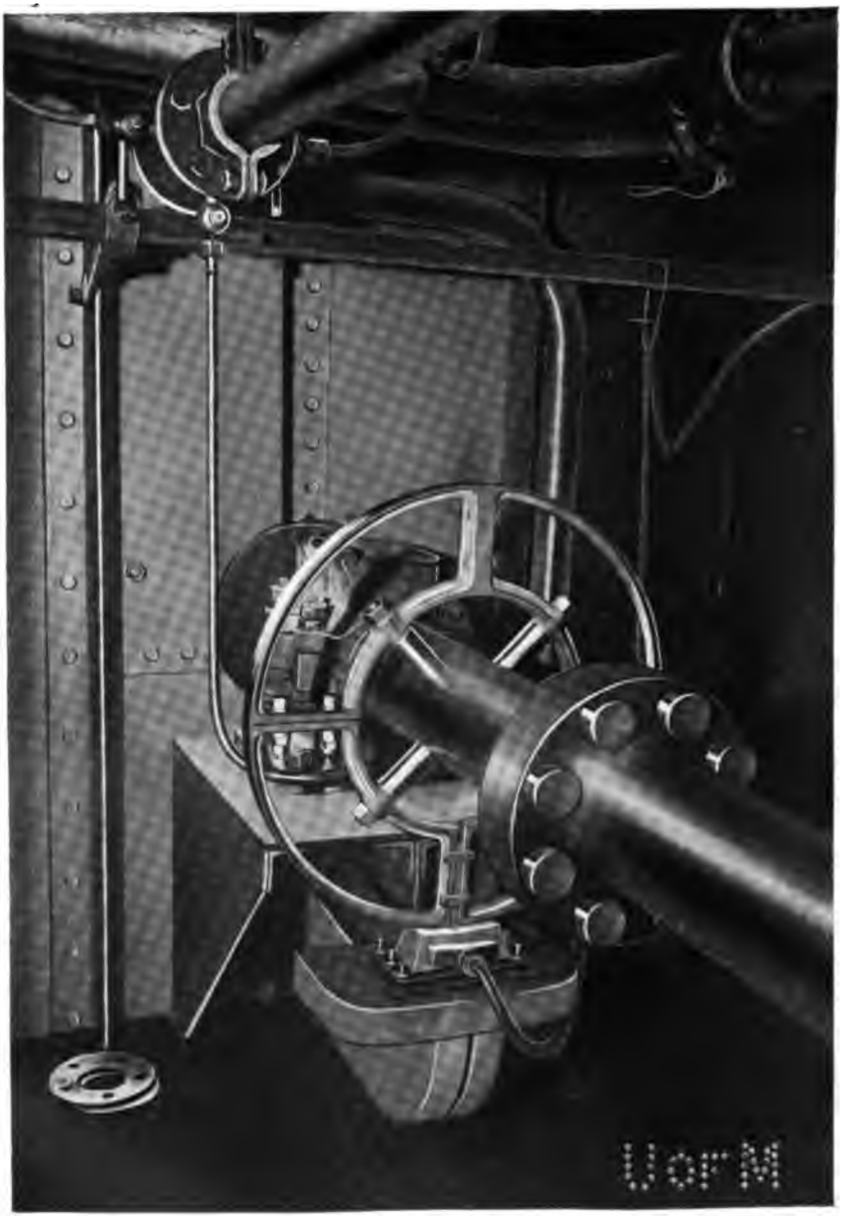
Contact arm B is then shifted from stud to stud, until the position of greatest silence in the receiver is once more obtained. When this position is found, the reading on the scale B, opposite the contact arm, represents the circumferential measurement of the angle of torsion of the shaft at the radius of the inductor windings. A current equal in strength at the same instant to that induced in the zero winding of inductor A is now being induced in that winding of inductor B which is in connection with the contact arm B; the scale reading thus represents the displacement of one magnet with regard to the other due to the torsion of the shaft. In the event of the torsion being found to be too great to be measured on scale B alone, contact arm A is shifted from stud to stud until a reading can be obtained on scale B, the torsion reading being equal to the sum of the readings on scales A and B. The reading corresponding to any large displacement of one magnet relatively to the other is thus easily obtained by the combined use of the scales A and B.

Type A Instruments.

This type has been designed for measuring the torsion of shafts having uniform turning moments, *e.g.*, shafting driven by turbines, electric motors, &c. &c.

To Fix the Apparatus.—A set of instruments as applied to a shaft consist of the following items:—(1) Two light gun-metal wheels, each fitted with a permanent magnet; (2) two inductors; and (3) a recording box. Before fixing up the apparatus it is first necessary to determine what length of the available shafting will be required to give a reasonably large torsional reading. The power to be transmitted by, and the revolutions of, the shaft at the maximum load are first approximately calculated. The diameter of the shaft is then measured, and the inductor constant noted. (This constant is the actual radius in inches at which the torsion of the shaft is measured; and it is stamped on all inductors.) Having made these observations, the length of shafting in feet required to give approximately 1 in. of reading is obtained by solving the formula,

$$\frac{1.53 \times R \times d^4}{C \times H.P.} = \text{Length.}$$



Torsion Meter, Shaft Wheel, Magnet, and Inductor.
Denny-Johnson Torsion Meter.

[*To face page* 264.

Where R = revolutions per minute of the shaft, C = inductor constant, d = diameter of shaft in inches, H.P. = horse-power (approximate), and 1.53 is a constant which takes account of the figure 140, a constant which has been ascertained from a great number of static calibrations of shafts, and which is an expression of the torsional resistance of solid iron and steel shafting. In the case of hollow shafting the torsional strength can best be found by calibrating the shaft (see page 267), but it is probable that the torsional strength is the same as that of a solid shaft of the same diameter minus the resistance due to a solid shaft which would fill the bore, and, in the absence of calibration, the error due to this assumption is probably a negligible quantity.

Having found the length of shafting which will give 1 in. of reading, this length is selected if available, and the wheels fixed to the shaft. It is desirable to select the greatest length of shafting available, provided the length chosen is such that the full load reading shall not exceed 1 in., it being advisable to have the remainder of the scale in reserve in the event of the power, &c., having been underestimated. In most cases it will be advisable to sacrifice a short length of the available shafting in order that the wheels may be fixed close to bearings (as shown in the illustration), thus allowing the wood or iron soles on which the inductors are to be mounted to be bolted to the bearing stools. The advantage of this is that any vibration that may be present when the shaft is running affects the shaft and bearing stools equally. If necessary, however, the inductors may be fixed in any other position, provided that a good solid base is provided. After fixing the wheels in place, the inductors should be screwed to the soles, and set concentrically with the wheels. The setting is done by means of two setting pieces supplied with the instrument. These are inserted between the top of the inductor and its wheel, the levelling screws in the base of the inductor being adjusted until the setting pieces both fit tightly and bear throughout their entire length. The locking nuts in the base are then tightened up and the setting pieces removed. The top or covering plate of each inductor has two lines or grooves cut across it, and the wheel in connection with the inductor fixed at the turbine or engine end of the shaft must be adjusted on the shaft until the magnet on that wheel is directly above the line from which it will travel, and towards the other line when the shaft rotates. *It is necessary to set the magnet above the correct line (which is the zero line corresponding to the direction of rotation of the shaft) or no readings can be taken.* The magnet is arranged in a tight fitting groove in the wheel, so that by removing the bridge pieces which hold it in position the magnet may be lowered until its sharp edge rests exactly in the zero line. The wheel should then be clamped tight to the shaft, and the magnet raised and fixed in position with its edge about $\frac{1}{16}$ in. clear of the top of the inductor. A gauge-marked air gap is supplied for the purpose of testing the clearance.

In addition to the two zero lines there is also a line marked circumferentially on the top of the inductor, and when setting the inductor this line should be made to coincide with the lines which are cut on the magnet. The other inductor is set and fixed in a similar manner; but in this case the magnet must be set to the zero line at the opposite end of this inductor. The inductors must be so placed that the connections are on the same side, *i.e.*, both on the right or both on the left hand side of the inductors, in order that the induced currents of electricity may be in the proper directions, and the

trouble of altering the telephone connections thus avoided. The cables should now be fixed to the inductors. The plugs attached to the ends of the cables are inserted into the sockets on the inductors, care being taken to see that the numbers on the plug and socket carriers correspond. The plugs must all be so inserted that the arrows on the plug carriers always point from the lines to which the magnets are set and in the direction of the other line. (This is essential.) The cables are not the same length. The longer one is intended to go to the inductor farthest away from the recording box (care should therefore be exerted when fixing the inductors to see that the inductor intended for use with the long cable is placed at the proper end of the shaft). This will probably be the after-end in all ships. The recording box should be placed in a reasonably quiet place, preferably a room as far removed from the noise of the machinery as is possible. The cable connections are then made to the recording box, and as the plugs and sockets are numbered no mistake should be possible.

To Take a Reading.—First see that the *resistance* contact arms and the *main* contact arms in the recording box are both at zero. The telephone receiver is then placed to the ear of the observer, and when the shaft is revolving a ticking sound will be heard. Shift the *main* contact arm of the $\frac{1}{100}$ths scale from stud to stud until a position is found where no tick is heard, or where the sound is reduced to a minimum. The torsion reading will then be found on the scale opposite the *main* contact arm. Should the tick not cease entirely but appear to be equally low on each of two neighbouring studs, the correct position of the arm is between those studs. When no reduction of the ticking noise can be obtained by moving this arm, it is evident that the torsion of the shaft is greater than the range of the $\frac{1}{100}$ths scale. The arm of the $\frac{1}{5}$ths scale must then be shifted from stud to stud until a reading can be obtained on the $\frac{1}{100}$ths scale. The torsion reading will then be the sum of the two scale readings. We have assumed that no balancing or equalising of the currents by resistance was required, but in practice this will usually require to be done before readings can be accurately taken. The equalising is accomplished as follows:—First take a trial contact arms reading as described above, and then move one of the resistances (the correct one being found by trial) until a position is found where the sound in the receiver almost or entirely disappears. It may be necessary to slightly alter the trial reading, as well as the resistance, before silence is obtained, and in cases where the equalising cannot be satisfactorily accomplished this will no doubt be due to the receiver being too sensitive for the rate of revolution. This can be remedied by adjusting the tension screw at the back of the telephone receiver until the desired degree of sensitiveness is obtained. Having once found the correct position for the resistance contact arm, no further adjustment will be necessary for a particular shaft or set of shafts. When applying the instrument to another shaft, or in cases where the revolutions of the shaft under trial vary greatly, it may be desirable to slightly readjust the resistance contact arm and also the receiver. With turbine-driven shafting in ships, where the rate of revolution is sufficiently high, a very good check may be applied to the accuracy of the zero setting of the inductors in the following manner:—After the day's trial is over, let the steam be shut off one turbine, and while the ship is driving ahead with the remaining turbines note the reading of the idle shaft, which will continue to revolve due to the action of the water on its propeller. The friction of the shaft bearings and of the turbine

may be assumed to require a relatively negligible amount of power to be transmitted by the propeller along the shaft, so that the reading thus taken should agree with the zero reading. To find the zero reading by this check method, the main contact arms of the two scales are placed opposite the letters z, when no sound should then be heard in the receiver if the inductors have been properly set. If a sound be heard, the arm of the $\frac{1}{100}$ths scale must be shifted until the tick disappears, or until the sound is reduced to a minimum. If the reading thus obtained is less than the reading at z, then

Recording Box.

the difference between the two readings is the amount of zero error, and must be added to the whole of the readings previously taken. If, on the other hand, the reading is greater than that at z, then the difference between the two readings must be subtracted from the whole of the readings previously taken. The foregoing test need only be applied, however, in the event of doubtful readings being obtained, as the set zero will always be correct if the inductors are carefully set up.

To Calibrate the Shaft.—Previous to being placed in its position the

shaft should be calibrated. This is done by coupling the various lengths together to include the length over which the torsion is to be measured, and then setting it on bearers and twisting it by means of weights in the following manner:—Clamp one end of the shafting rigidly in position so that it cannot turn, and fix a stout lever to the other end. On to the same end of the shaft fix a pointer 6 or 7 ft. long, so that it will move over a paper scale. Weights should then be attached to the outer end of the lever, and the deflection of the pointer in inches noted as the weight is increased until a set of readings from zero up to the maximum calculated torque are obtained. A curve is then drawn, the ordinates being foot-pounds and the abscissal torsion in inches.

To Find the Shaft H.P. from the Torsion Reading.—From the calibration curve the foot-pounds corresponding to any torsional reading of the shaft taken by the torsion meter may be easily found. The torsion meter readings must first of course be made to correspond with the length of calibrated shaft and to the calibration radius by means of the formula,

$$\frac{r \times l \times L^1}{C \times L} = K,$$

where r = torsion meter reading in inches, l = length of pointer in inches, L^1 = length in feet of calibrated shaft, C = inductor constant, and L = length in feet of shaft from which the torsion meter reading is obtained, K = reading of torsion meter corresponding to the curve.

Having found from the curve the foot-pounds corresponding to any reading of the torsion meter, the horse-power transmitted is then found by means of the formula,

$$\frac{F \times R}{5255} = \text{horse-power transmitted by the shaft,}$$

where F = foot-pounds, R = revolutions per minute of the shaft, and 5255 is a constant.

In the case of solid shafts of iron or steel, instead of calibrating the shaft the constant 140 (which figure represents the average torsional resistance of iron and steel shafts as found from numerous calibrations of shafts) may be accepted as correct, and the horse-power transmitted by the shaft found by means of the formula,

$$\frac{1.53 \times r \times d^4 \times R}{C \times L} = \text{horse-power.}$$

In the above formula, d = diameter in inches of the shaft, R = revolutions per minute of the shaft, C = inductor constant, L = length in feet of shaft, r = reading of torsion meter in inches, and 1.53 is a constant.

Hollow shafting should be calibrated, as the torsional resistance varies with the bore, which may not be sufficiently uniform to permit of very accurate calculation.

It is strongly recommended, however, that all shafting, whether hollow or solid, should be calibrated if at all possible, as very great accuracy is then assured. When the constant 140 is used in place of calibration the average possible error is about 1 per cent.

Recent Improvements in Turbine Construction and Design.

Blading.—The blading now generally adopted is that of the

"segment" type, which allows of the simultaneous construction of the rotors and casings with the blade fitting, as the various segments of blades and "packers" only require to be placed in position and caulked down into the grooves to complete the work. This effects a considerable saving of time in the work of construction. The Admiralty practice of thinning away the blade tips to prevent damage should wear-down occur is usually carried out in all long blades at least, and often throughout the blading.

Wheels.—In large turbines the wheels are now usually made of steel forgings in place of steel castings as formerly. When large wheels are of cast steel special allowance is made for shrinkage by having a cut made in the rim, which is afterwards filled in by a fitting strip when the metal has cooled down and set. This obviates the setting up of severe strains due to the casting cooling and contracting unequally owing to varying thickness, &c. Brass wheels are also being experimented with.

Glands.—Various methods of gland fittings have been experimented with during the past two years, and the type now adopted, and evidently standardised, consists of four Ramsbottom rings in combination with twenty or more radial fins, the gland case itself being in two halves. In the H.P. turbines steam is admitted to a pocket between the fins and rings during heating up, but when running the same pocket acts as a "leak-off" to the 3rd expansion of the L.P. turbines.

In the L.P. turbines the pockets referred to admit steam at a low pressure, and do not act as "leak-offs," as these are not required with L.P. turbines.

A type of gland made up and fitted in three segments has been fitted in a few cases (notably in the turbines of the T.SS. "Cairo" and "Heliopolis"), but owing to the labour and time required in overhauling and putting together again, this type has wisely been discarded, the arrangement being much too complicated for practical operation. It may be stated that the problem of the ideal gland has yet to be solved, as the present arrangements do not give complete satisfaction.

Dummies.—The dummy rings are now much more undercut than formerly, resulting in a reduced thickness of metal presented for contact should longitudinal wear or displacement of the rotor take place. This tends to less risk of serious damage resulting from contact of the grooves and rings.

In a few cases which came under the observation of the writer the dummies had been in actual contact when running, but no serious damage ensued, the only evidence of metallic contact being deep circumferential marking of the grooves in the dummy piston. It is also worthy of note that the dummy "leak-offs" are now abandoned, as it is considered that they merely form a passage through which

steam will find its way unnecessarily from one point of the turbine to another. As trouble has been experienced by corrosion occurring in the dummies, due to galvanic action between the brass rings and cast-iron case, it has been suggested to make the dummy casings and pistons all of brass instead of cast iron, but the cost of this material forms an important objection, and, in any case, it is likely the corrosion would still take place at some other position.

Oil and Water Service.—The oil service is now much improved in many ways, the oil circulating through sight glasses fixed to the bearings, which afford a reliable index to the oil flow. In some cases small cocks are fitted at different positions on the bearings, and these have a small hole drilled through the shell so that when the plug is turned round the oil spurts out if present. In the most recent system the oil is cooled independently before entering the bearings, the water service to the bearings having been done away with. The oil is also filtered in special gravity tanks before or after cooling, and to allow for temporary stoppage of the oil flow from the oil pumps the " reserve tank " system has been adopted in some cases, but cannot be said to be altogether satisfactory, as the reserve supply of oil in the overhead tanks would only suffice for a very few minutes' actual running of the turbines should the pump supply be cut off.

Running and Upkeep of Turbines.

An important point in favour of turbines is the small amount of repairs required, as owing to the small number of bearings and moving parts, and there being practically no wear of bearings, repairs are reduced to a minimum. To attain this result and to give best economy in running turbines, the following advice may be of use. In dealing with the bearings, care should be taken that all the oil supplied to bearings is carefully strained and filtered. In most of ships filters are fitted on the supply pipe to bearings. The filtering cloths in these filters should be kept as fine as possible, and to ensure quickness in overhauling, a spare set of filtering grids should be kept. The filter is opened, and dirty grids taken out, and clean ones put in. The dirty ones can now be thoroughly cleaned, ready for reinstatement. The oil should be kept at as steady a temperature as possible, and oil in suction tank kept up to requisite level.

A wear-down gauge is usually supplied similar in purpose as in a reciprocating engine. This should be used frequently to ascertain if any wear has taken place in bearings.

The adjusting or thrust block should be examined to see that there has been no wear on the rings, and if so, then it is necessary to readjust same in relation to the rotor dummy. It will be seen that the finer the dummy adjustment the more economical the turbines will be. It is not advisable to bring dummy too close, as disaster may occur, but a dummy can be run at .01 clear of blades on casing dummy, and this is about as close as it is safe to go. When the

clearance is fined down to this extent, it is advisable to do this setting after turbine is warmed up. Previous to heating up, the rotor should be pressed aft until the dummy is .030 clear of blades. After turbine is thoroughly warmed up, the rotor can be brought forward again and clearance fined down to .01 or .015 as is thought safe. This is ascertained from the finger piece. The clearance being fine, less steam will escape to the interior of the rotor, instead of doing work going through the blades. Turbines are designed so that the propeller thrust will be balanced by the steam thrust on the blades, and if this balance is perfect, then the shaft collars will not touch the rings on the adjusting bush, but will be floating clear. After machinery is stopped and cooling down, it is advisable, if working with a fine dummy clearance, to press the rotor aft, until dummy is from .030 to .040 clear, as in the event of shaft cooling down before rotor, it will draw rotor forward and considerably decrease the dummy clearance. This operation can be avoided by keeping a gentle flow of steam acting on the shaft, at the forward gland, so as to allow the rotor and casing to cool before shaft is cooled. In running turbines, care must be taken to avoid priming of boilers. This is a very important matter, as the passing of boiler water into the interior of the turbines may not do any material damage to the blading at the time, but if the boiler water is salt, then salt deposit will take place, and this, becoming solidified, will throw the rotor out of balance. Salt and other sediment being carried into the turbines will get in between the blades, and may result in closing up the blading in the 1st and 2nd expansions, with the result that the steam will not be doing any effective work, but will flow past the tip of the blades, and cause a higher pressure to show in the receiver. When starting up, the drains on the H.P. casing should be open, and the air pump drawing from bottom of L.P. casing so as to clear any water that may gather, before the turbines are thoroughly heated up.

Attention should be given to revolutions of each turbine, and these should be made to correspond as nearly as possible. It may be found that one L.P. is running faster than the other; in this event, the valve on the L.P. main steam inlet of the turbine which is running fastest should be closed a little so as to cause an increased amount of steam to pass to the turbine which is running slowest. The turbine casings are lifted for survey by Board of Trade once every twelve months in passenger ships, and a thorough examination should then be made of blading to see that no fraying of blades is taking place, and of interior of rotor drums to see that no corrosion or salting up is taking place.

Combined Reciprocating Engines and Turbines. — The most recent practice in mercantile steamers is the combination of reciprocating engines and turbines for vessels of moderate or low speeds. Several steamers of this class are at present under construction, and the arrangement consists of two wing triple or quadruple expansion

engines exhausting into a central low pressure ahead turbine, driving a third shaft and propeller, the revolution speed of the centre turbine shaft being much higher than that of the wing shafts. An alternative design is that of two wing low pressure turbines and one centre reciprocating engine. The turbine for the Dominion Line steamer now under construction by Messrs Harland & Wolff is 10 ft. 4 in. in diameter over blades, and is supplied with 6 expansions, the blade heights varying from 3 in. to 10 in.; while the turbine for the N.Z. Shipping Company steamer has a diameter of about 7 ft. 6 in. and is fitted with 5 expansions, 3 of increasing blade height and 2 of uniform height.

In the steamers mentioned the engines are arranged to be run as follows:—

(1) Boiler steam to both H.P. cylinders, and exhaust from these to turbine, the exhaust from the turbine being divided and led into two separate condensers.

(2) Boiler steam to both H.P. cylinders, and exhaust from these to condensers direct, the turbine being then cut off. This is required when running astern as the turbine is for ahead running only, and may be used for ahead running with two propellers only.

(3) Boiler steam to either H.P. cylinder, and exhaust from L.P. to centre turbine, then into one condenser only.

These combinations are obtained by the use of "change valves" fitted on the L.P. exhaust pipes, and by large butterfly valves fitted in the turbine exhaust branches. The change valves admit the reciprocating exhaust steam of either side to the turbine, or to the condenser as required, and the large valves in the turbine exhaust pipe shut off the condenser on either side as may become necessary should one reciprocating engine require to be disconnected through breakdown.

Benefits of the System.—As, broadly speaking, the economy of the reciprocating engine depends chiefly on high-pressure steam, and the turbine on low-pressure steam, the judicious combination of the two ought to result in higher efficiency results. The theoretical heat gain may be figured out as follows:—

L.P. terminal pressure = 8 lbs. absolute.
Condenser back pressure = 2 „
Assume dryness fraction at 8 lbs. absolute = .82.
„ „ „ 2 „ = .78.

Then on referring to the steam table, page 3, the latent heat at 8 lbs. = 985.7 B.T.U., and the latent heat at 2 lbs. pressure = 1025.8 B.T.U. The temperature at 8 lbs. = 182.9° Fahr., and at 2 lbs. 126.1° Fahr., and adding 461 for absolute temperature to each of these we obtain $182.9 + 461 = 643.9$ and $126.3 + 461 = 587.3$.

Then referring to page 34 we find the formula for work done by adiabatic expansion.

Therefore, $985.7 \times .82 - 1025.8 \times .78 + 643.9 - 587.3$
$= 808.274 - 800.124 + 643.9 - 587.3$
$= 808.274 + 643.9 - 800.124 - 587.3$
$= 1452.174 - 1387.42 = 64.75$ B.T.U. per lb. steam.

Assuming a steam flow of, say, 1882 lbs. per minute, and an efficiency of 55 per cent. for low-pressure turbines—

Then, Equivalent I.H.P. $= \left.\dfrac{64.75 \times 1882 \times 778 \times .55}{33000}\right\} = 1580$ (nearly) horse-power.

The turbine is therefore most effective in dealing with steam of a pressure which cannot be utilised with benefit in a triple or quadruple expansion engine, owing more particularly to the huge volumes involved, and requiring increase of weight, space, and frictional losses.

It is understood that the L.P. exhaust pressure to the centre turbine will not be more than 7 or 8 lbs. absolute. This will produce a difference in the usual L.P. cylinder diagram cards, bringing up the exhaust line to a position much nearer the atmospheric line than usual.

The loss of work energy so represented by the reduced indicator card area in the L.P. engine will be more than balanced by the increase of power developed in the turbine.

The economical result of the combination arrangement is, to all appearance, beyond question, and may in time, with suitable improvements which experience suggests, prove adaptable for the usual tramp steamer speed of from 8 to 10 or 11 knots.

An innovation has been made in the case of the turbine glands, which, instead of the frictionless steam packing hitherto adopted, have in one case been changed for the usual marine type of piston rod gland, consisting of rack and pinion screwing-up gear with soft packing inside, this type of gland being fitted at both ends of the turbine.

Theoretical Data for Combination System of Reciprocating Engines and Turbines.

H.P. initial pressure - -	185 lbs. (gauge).
Turbine initial pressure - -	12 in. vacuum, or 9 lbs. absolute.
Condenser - - - - -	28 in. vacuum, or 1 lb. absolute.
Steam flow per hour - -	52,000 lbs.
Reciprocating engine over-all efficiency (assumed) - -	.75.
Turbine over-all efficiency (assumed) - - - -	.54.

Then,

$185 + 15 = 200$ absolute initial H.P. engine pressure.

200 lbs. $= 381.7°$ temp. and $381.7° + 461° = 842.7°$ absolute temp.

200 lbs. $= 845$ B.T.U. of latent heat.

9 lbs. absolute $= 188.3°$ temp. and $188.3 + 461 = 649.3$.

9 lbs. absolute $= 981.9$ B.T.U. of latent heat.

1 lb. absolute $= 102.1°$ temp. and $102.1 + 461 = 563.1$.

1 lb. absolute $= 1042.9$ B.T.U. of latent heat.

Dryness fraction : 1 for initial pressure of engine, .845 for initial pressure of turbine, and .771 for terminal pressure of turbine.

Then, heat drop per lb. steam in reciprocating engine working between 200 lbs. initial and 9 lbs. terminal pressure, and assuming adiabatic expansion throughout

$$= 845 \times 1 - 981.9 \times .845 + 842.7 - 649.3$$
$$= 845 - 829.70 + 842.7 - 649.3 = 208.7 \text{ B.T.U.}$$

Heat drop per lb. steam in turbine working between 9 lbs. initial pressure and 1 lb. terminal pressure, also adiabatic expansion,

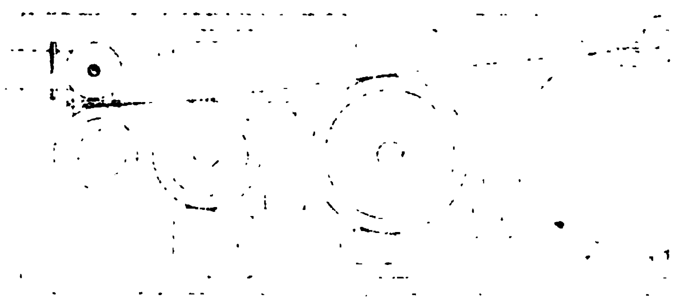
$$981.9 \times .845 - 1042.9 \times .771 + 649.3 - 563.1 = 111.9 \text{ B.T.U.}$$

Steam flow per hour through each reciprocating engine	$= 52{,}000 \div 2 = 26{,}000$ lbs.
Horse-power developed in each engine	$= \dfrac{26000 \times 208.7 \times 778 \times .75}{60 \times 33000} = 1599.$
Total horse-power developed in both engines	$= 1599 \times 2 = 3198.$
Steam flow per hour through turbine	$= 52{,}000$ lbs.
Horse-power developed in turbine	$= \dfrac{52000 \times 111.9 \times 778 \times .54}{60 \times 33000} = 1234.$
Combined horse-power	$= 3198 + 1234 = 4432.$
Lbs. steam per horse-power hour	$= 52000 \div 4432 = 11.7$ lbs.
Lbs. coal per horse-power hour (evaporation 9 lbs.)	$= 11.7 \div 9 = 1.3$ lbs.

Appended will be found an extract of a paper by the Hon. C. A. Parsons and R. J. Walker read before the Institution of Naval Architects, 9th April 1908, entitled "The Combination System of Reciprocating Engines and Steam Turbines," and in which various data are given relating to the power and consumption expected by this combined arrangement:—

"It may be said that perhaps the most important field for the combined system of machinery as applied to marine propulsion is for those installations where the designed full speed of the vessel falls below the range suitable for an all-turbine arrangement, the reciprocating engine working in the region of pressure drop where the conditions are best suited for it, and the turbine utilising that portion of the expansion diagram which the reciprocating engine is not able to utilise efficiently. It is generally well known that an all-turbine arrangement has not been advocated by us for ships where the designed speed falls below 15 or 16 knots, excepting in some special cases, such as yachts ; and for vessels of moderate or slow speed the combination system of machinery appears to be eminently suitable.

"In a good quadruple reciprocating engine, the steam is expanded down to the pressure of release, about 10 lbs. absolute, and gains in economy as the vacuum is increased up to about 25 in. or 26 in. ;



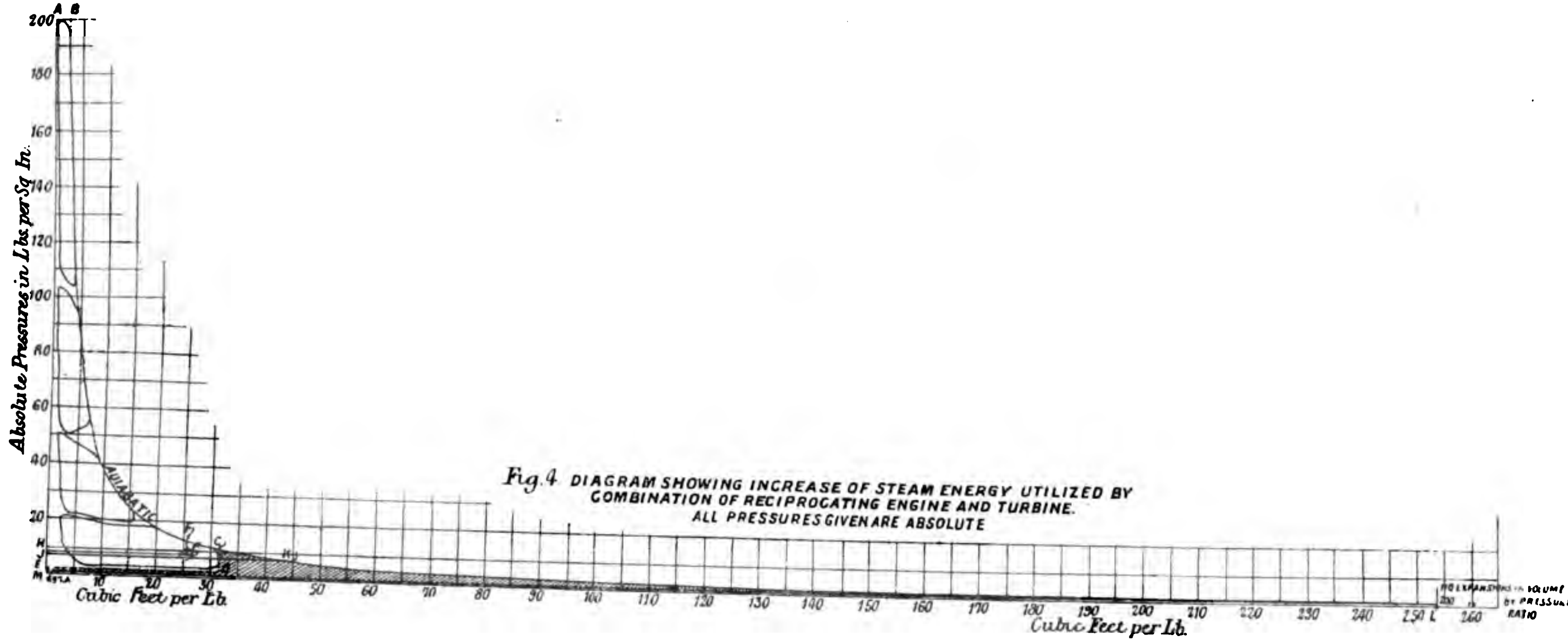


Fig. 4 DIAGRAM SHOWING INCREASE OF STEAM ENERGY UTILIZED BY COMBINATION OF RECIPROCATING ENGINE AND TURBINE.
ALL PRESSURES GIVEN ARE ABSOLUTE

whereas in a turbine it is possible to deal economically with very low-pressure steam, and to expand this low-pressure steam to a low absolute pressure corresponding to the highest vacuum obtainable in turbine practice.

"Fig. 4 shows in diagrammatic form the advantage of the combined system of machinery. The total area of the diagram represents the maximum energy that could be obtained, theoretically, from the steam if it were expanded down to the pressure in the condenser. The area enclosed by lines A, B, C, D, and E shows the theoretical maximum energy realisable in a quadruple engine from 200 lbs. pressure to 26 in. vacuum, and the area cross-hatched the additional energy that can be utilised in a turbine, but which cannot be economically used in a reciprocating engine.

"In a combination system, the most suitable initial pressure for the turbine, or the dividing line between the reciprocating engine and the turbine, will greatly depend upon the conditions of service of the particular vessel taken. The reciprocating engine, or engines, could be designed to exhaust at a pressure of between 8 lbs. and 16 lbs. absolute, or even at a slightly higher pressure, if necessary, to meet the conditions required. From an estimate of the theoretical efficiency under the various conditions of pressure, as set forth in the following table, it would appear, apart from any practical considerations, that there is nothing to choose between an initial pressure at the turbine of between 7 lbs. and 15 lbs. absolute, any pressure within this limit appearing to give the most economical result.

	Initial Pressure Turbine.	Reciprocating Engine Back Pressure.	Theoretical B.T.U. per lb. of Steam.		
			R.E.	Turbine	Total.
Assuming 200 lbs. absolute at reciprocating engine to 28 in. vacuum at condenser. 1 lb. = .77 dryness	15	16	178	142	320
	12½	13½	189	131	320
	7	8	218	100	318

"Table Explanatory of Diagram (Fig. 4), showing Increase of Steam Energy Utilised by Combination of Reciprocating Engine and Turbine. (All Pressures given are Absolute.)

Quadruple Expansion Reciprocating Engine Exhausting to Condenser Direct.

Area enclosed by A, B, C, D, E = maximum energy realisable in quadruple engine, from 200 lbs. to 26 in. vacuum, with point of release at 10 lbs. = 256 B.T.U.

Area cross-hatched represents energy which reciprocating engine cannot efficiently utilise, but which can be used in a turbine = 73 B.T.U.

Combination of Triple Expansion Reciprocating Engine Exhausting to Turbine and thence to Condenser.

Area enclosed by A, B, F, G, H = maximum energy realisable in triple engine from 200 lbs. to 8 lbs., with point of release at 13 lbs. = 219 B.T.U.

Area enclosed by J, K, L, M = energy available for turbine from 7 lbs. to 28 in. vacuum, receiving wet steam from reciprocating engine = 100 B.T.U.

Total energy of combination = 319 B.T.U.

Theoretically, total energy of combination is 24½ per cent. greater than that of quadruple engine.

It is estimated that a large portion of this additional energy can be expected to be realised by the combined system in the shape of increased power to drive the vessel, or, on the other hand, increased economy.

The theoretical figures are computed on the basis of adiabatic expansion throughout.

"In the case of a vessel which runs on service continually at or about her designed full speed, an initial pressure of about 7 lbs. absolute at the turbine appears most suitable. In a vessel which does part of her running at the designed power, and part at a considerably reduced power, it is desirable to design the turbines so that the initial pressure would not fall below 7 lbs. absolute when running under the lower conditions of power.

"It may be of interest at this stage to consider the disposition of the turbines in combination with reciprocating engines on board ship. The arrangement of the turbine, or turbines, depends greatly whether the vessel is to be fitted with single or twin-screw reciprocating engines. With a single reciprocating engine, one turbine, two turbines in 'series,' or two turbines in 'parallel' could be fitted, each turbine driving a separate shaft in addition to the reciprocator shaft. With twin-screw reciprocating engines, an arrangement of one turbine in the centre of the vessel, two turbines in 'parallel,' or two turbines in 'series,' could be adopted. The arrangement which seems to commend itself generally to shipowners and builders, where twin-screw reciprocating engines are fitted, is the arrangement with the turbine on the centre shaft.

"In the combination proposals set forth in columns B and C in the following table, it may be mentioned that in this particular inquiry the shipowners wished to have the advantage of the additional power and increase in speed of the vessel on the same coal consumption as for the twin quadruple engines. In some instances an increase in speed might not be desired, in which case the boilers and engines could be

reduced in size by the estimated amount of saving in consumption, so that the total indicated horse-power of the combination did not exceed that required with twin quadruple engines. This would considerably reduce the total weight of machinery, and also the bunker capacity for

	A	B	C
	Twin Quadruple Reciprocating Engines.	Three-Cylinder Triple-Expansion Twin Reciprocating Engines, with Two Low-Pressure Turbines in Parallel.	Four-Cylinder Triple-Expansion Twin Reciprocating Engines, with One Low-Pressure Turbine.
Dimensions of reciprocating engines	25, 36½, 51½, 75 55	27, 42, 66 48	26, 39, 46, 46 42
Revolutions of reciprocating engines	84	85	100
Piston speed of reciprocating engines	770	680	700
Boiler pressure	213 lbs.	213 lbs.	213 lbs.
Estimated pressure at H.P. receiver	200 lbs.	200 lbs.	200 lbs.
Initial pressure turbines	..	7 lbs. absolute	7 lbs. absolute
Vacuum in condenser	26 in.	28 in.	28 in.
Steam consumption, main engines only	95,000 lbs. per hour	95,000 lbs. per hour	95,000 lbs. per hour
I.H.P. reciprocating engines	7,300	6,300	6,300
Estimated equivalent I.H.P. of turbines	...	2,000	2,000
Total I.H.P.	7,300	8,300	8,300
Per cent. increase power	...	13.7 p. c.	13.7 p. c.
Estimated speed	15.5 knots	16.2 knots	16.2 knots
Steam consumption, lbs. per total I.H.P. per hour main engines)	13 lbs.	11.45 lbs.	11.45 lbs.
Steaming weight of engines and boilers (reciprocating engines)	...	1,430 tons	1,455 tons
Weight of turbine installation	...	65 "	70 "
Total steaming weight	1,560 tons	1,495 "	1,525 "
Revolutions of turbines	...	480	320

a given distance. This saving in the weight of the machinery and in the bunkers would enable the vessel to carry an equivalent addition in dead-weight cargo. Then, again, if we take the indicated horse-power of 8,300 for the combination, and assume that quadruple

engines and boilers were required to give an equivalent power, the extra total weight of machinery would be, roughly, 160 tons, in addition to an increase of about 12 per cent. in coal consumption for the same power.

"Suitable arrangements are made for changing the flow of steam of the low-pressure cylinder exhaust of the reciprocating engine from the turbine to the condenser. This can be done in two or three ways. One method is to have an ordinary change valve of the piston type, or ordinary double-beat spring-loaded valve actuated by links connected to the weigh-shaft of the main engine, which would automatically change the flow of steam to the condenser when the engine was reversed. With this arrangement, when going ahead on one side of the ship, the steam from the reciprocating engine would flow through the turbine, but there does not appear to be any objection to this even if we consider the twin-screw reciprocating arrangement with a single turbine on the centre shaft. It might be rather an advantage, than otherwise, to allow the steam from the engine going ahead to pass through the turbine, as the centre propeller revolving would accelerate the feed of water on the rudder, and augment the turning power of the vessel.

"Another method would be to work these valves independent of the main engines, actuated by an hydraulic engine, or by an ordinary steam-driven reversing engine. With this arrangement, the low-pressure turbine would be cut out altogether and the reciprocating engine would exhaust to the condenser whether going ahead or astern during manœuvring."

Turbine Boilers, Condensers, and Auxiliary Machinery.

Most of turbine-driven ships in the merchant service have boilers of the usual Scotch type, but in the Navy the water-tube boiler has been largely adopted. These are usually of the Yarrow or Babcock & Wilcox type. The Scotch boilers are usually fitted with the Howden system of forced draught, but with the water-tube boiler the closed stokehold principle is in use. In the Navy liquid fuel has also been adopted—principally in high-speed coastal destroyers—and has given good results. Turbine condensers differ but little from the condenser of an ordinary reciprocating engine, but the connection for the exhaust pipe from the turbine is of extra large area, as owing to the low terminal pressure the volume of steam is excessive. Baffle plates, consisting of steel or bronze plates perforated with holes, are fitted in the interior of the condenser, and these tend to baffle the passage of the steam, and so use the cooling surface to the best extent. It is important in turbine installation to maintain a high vacuum, and to attain this result large air and circulating pumps are fitted. The circulating pumps are of the centrifugal type, and are driven by a separate engine. The air pumps are of the usual bucket type, and are also self-driven. In some ships "dry air"

pumps are fitted; these are sometimes driven by the main air pump engine.

The purpose of the dry air pump is to draw the air only from the condenser, leaving the wet air pump to deal with the water, thus increasing the vacuum in the condenser. The dry air pump is usually put to work when the turbines are running at full speed; whilst manœuvring the wet air pumps only are used.

Pumps are fitted for the forced lubrication service, and there are usually two of them—one acts as spare, and can be brought into use whilst the other is being overhauled. Feed and bilge pumps are also driven independently, as owing to the high running speed of the turbines, it has not been found practicable to drive any of the auxiliary machinery from them.

PROBLEMS IN TURBINE DESIGN

(FOR JUNIOR STUDENTS).

(*See Appendix, page* 324, *for General Notes.*)

Steam and Blade Speeds.

RULE.—In marine practice, the ratio of V_t to V_s varies from about .31 to .52.

Therefore, (1) $V_t \div V_s = \text{Ratio } V_t, V_s$

„ (2) $V_s \times \text{Ratio} = V_t$

„ (3) $V_t \div \text{Ratio} = V_s$

Where, V_t = Peripheral Blade Speed.

„ V_s = Steam Speed.

EXAMPLE.—Determine the required mean blade velocity if the ratio V_t to V_s is .45 and the steam speed is to be 200 ft. per sec.

Then, $200 \times .45 = 90$ ft. per sec.

NOTE.—Blade speed per sec. $\times 60 \div$ Revs. $\times 3.1416$ = Diameter of rotor, across blades (mean).

1. Calculate the mean blade velocity if the steam speed is 300 ft. per sec. and the ratio $V_t \div V^s = .4$. Answer. 120 ft. per sec.

2. What is the necessary steam speed per sec. for a blade velocity of 90 ft. per sec. if the ratio $V_t \div V_s$ is equal to .45? Answer. 200 ft. per sec.

3. The ratio of V_t to V_s is .48, and the steam speed 250 ft. per sec. Find the mean blade velocity. Answer. 120 ft. per sec

4. If the steam speed is 350 ft. per sec., and the blade velocity 150 ft. per sec., calculate the ratio of V_t to V_s. Answer. .42 ratio.

Diameter of Rotors, &c.

RULE.—Rotor surface velocity per minute = Rotor diam. × 3.1416 × Revs.

Therefore, $$\frac{\text{Velocity per min.}}{3.1416 \times \text{Revs.}} = \text{Diam. of rotor.}$$

and, $$\frac{\text{Velocity}}{\text{Diam.} \times 3.1416} = \text{Revolutions.}$$

NOTE.—This rule neglects the difference in diameter due to the blade heights.

EXAMPLE. — Determine the required diameter of rotor, if the surface velocity is to be 110 ft. per sec. and the revolutions 250 per min.

$$\text{Diam. across blades} = \frac{\text{Blade velocity per min.}}{3.1416 \times \text{Revs.}} = \frac{110 \times 60}{3.1416 \times 250} = 8.4 \text{ ft.}$$

1. Calculate the required diameter of L.P. rotor for a velocity of 9,000 ft. per min., and revolutions of 400 per min. Answer. 7.16 ft. diameter.

2. Calculate the required diameter of H.P. rotor if the velocity is 6,000 ft. per min., and revolutions of 550 per min. Answer. 3.47 ft. diameter.

3. Calculate the required diameter of reverse rotor for a velocity of 5,000 ft. per min., and revolutions of 400 per min. Answer. 3.97 ft. diameter.

4. Determine the required revolutions for a turbine if the velocity is 8,000 ft. per min., and rotor diameter 6 ft. Answer. 424 revolutions.

5. Find the revolutions to correspond with a velocity of 9,000 ft. per min., and rotor diameter of 7 ft. Answer. 409 revolutions.

6. Calculate the necessary revolutions for a velocity of 100 ft. per sec., and rotor diameter of 5 ft. 6 in. Answer. 347 revolutions.

7. Find the revolutions for a velocity of 90 ft. per sec., and rotor diameter of 4 ft. Answer. 429 revolutions.

8. Find the revolutions required for a velocity of 160 ft. per sec., and rotor diameter of 14 ft. Answer. 218 revolutions.

Steam Velocities and Heat Drops.

RULE.—British thermal units × 778 = Foot lbs. of energy given up.
and, Foot lbs. ÷ 778 = B.T.U. heat drop.

Again, $$\frac{W \times V^2}{64.4} = \text{Foot lbs. of energy in first guide blades.}$$

Where W = Weight in Pounds,
V = Velocity of Steam in feet per second.

Also, $$\frac{(V^2 - v^2) \times W}{64.4} = \text{Foot lbs. given up in any moving blades.}$$

Therefore, $$64.4 \times \text{Foot lbs.} = W \times V^2$$

and, $$\sqrt{\frac{64.4 \times \text{Foot lbs.}}{W}} = V \text{ (Guide blades),}$$

or, $$\sqrt{\frac{64.4 \times \text{Foot lbs.} + v^2}{W}} = V \text{ (Moving blades).}$$

Where V = Velocity of Steam in feet per second at exit edge of blades.
v = Velocity of Steam in feet per second at admission edge of blades.
W = Weight of Steam in pounds.
$64.4 = 32.2 \times 2$ (acceleration due to gravity per sec. per sec.).

EXAMPLE (1).—If the initial velocity of the steam is 220 ft. per sec. and the exit velocity 300 ft. per sec., calculate the kinetic energy given out in ft. lbs. per lb. of steam supplied (neglecting losses). Also determine the heat drop in B.T.U.

Then, $$\frac{(V^2 - V^2) \times W}{64.4} = \frac{(300^2 - 220^2) \times 1}{64.4} = 645.9$$

and, Heat drop $= 645.9 \div 778 = .83$ B.T.U.

EXAMPLE (2).—The initial steam velocity is 130 ft. per sec., and the heat drop per lb. of steam supplied 1.8 units. Determine (1) the required exit velocity, and (2) the kinetic energy in ft. lbs. given out per lb. steam supplied (frictional and leakage losses neglected).

Then, (1) $$V = \sqrt{\frac{64.4 \times \text{Foot lbs.} + V^2}{W}}$$

$$= \sqrt{\frac{64.4 \times 1.8 \times 778 \times 130^2}{1}} = \sqrt{\frac{64.4 \times 1400.4 + 16900}{1}}$$

$$= \sqrt{\frac{90185.76 + 16900}{1}} = \sqrt{\frac{107085.76}{1}} = 327.5 \text{ ft. per sec.}$$

And, (2) Kinetic energy $= 1.8 \times 778 = 1400.4$ ft. lbs.

1. Calculate the work done in velocity acceleration by 1 lb. of steam in passing through the guide blades of a Parsons turbine at an initial velocity of 600 ft. per sec. Answer. 5,590 ft. lbs.

2. A lb. of steam enters the guide blades of a marine turbine at a velocity of 300 ft. per sec. Calculate (1) the work done and (2) the heat given up.

Answers. (1) 1,397 ft. lbs. (2) 1.79 B.T.U.

3. Calculate (1) the ft. lbs. of kinetic energy done in 1 lb. of steam when passing the first guide blades of a turbine at a velocity of 350 ft. per sec.; also (2) the B.T.U.

Answers. (1) 1,902 ft. lbs. (2) 2.44 B.T.U.

4. Calculate (1) the ft. lbs. of energy and (2) drop of heat which are produced by 1 lb. of steam at a velocity of 15,000 ft. per min.

Answers. (1) 970 ft. lbs. (2) 1.24 B.T.U.

5. One lb. of steam enters the moving blades of a marine turbine at a velocity of 200 ft. per sec., and leaves at a velocity of 300 ft. per sec. Calculate (1) the ft. lbs. of energy given out and (2) the heat drop.

Answers. (1) 776 ft. lbs. (2) .99 B.T.U.

6. Find the heat drop if the steam velocities are 300 ft. and 400 ft. per sec. Answer. 1.39 B.T.U.

7. Find the required heat drop to give velocities of 300 ft. and 220 ft. per sec. in the moving blades. Answer. .82 B.T.U.

8. Calculate the steam velocity per sec. if the energy given out is 2,000 ft. lbs. for each lb. of steam supplied. Answer. 358 ft. per sec.

9. Find what velocity of steam is necessary to develop 1,000 ft. lbs. of energy per lb. of steam. Answer. 253 ft. per sec.

10. Twenty lbs. of steam passing through the guide blades of a turbine give out 10,000 ft. lbs. of work. Calculate the velocity of the steam. Answer. 179 ft. per sec.

11. One lb. of steam drops 2 B.T.U. in passing through the blades of a turbine. Calculate the steam velocity. Answer. 316 ft. per sec.

12. What steam velocity is required to obtain a heat drop of 2.5 B.T.U. per lb. of steam? Answer. 353 ft. per sec.

13. Calculate the exit velocity of the steam in the turbine blades if the ft. lbs. given out per lb. are 1,500 and the initial velocity is 200 ft. per sec. Answer. 369 ft. per sec.

14. The initial velocity is 300 ft. per sec., and the heat units given up 2. Calculate the steam velocity at the exhaust edge. Answer. 436 ft. per sec.

15. What exit velocity is necessary if the heat drop is 1.5 B.T.U., and the initial velocity 260 ft. per sec.? Answer. 377 ft. per sec.

16. Heat drop 3 B.T.U., initial velocity 6,000 ft. per min. Calculate the exit velocity. Answer. 400 ft. per sec.

17. One lb. of steam contains 1,200 B.T.U. on entering the blades and 1,196 B.T.U. on leaving the blades of a turbine. Calculate the exit velocity if the admission velocity is 300 ft. per sec. Answer. 538 ft. per sec.

18. Calculate the exit velocity of the steam if the initial velocity is 160 ft. per sec., and the heat drop 1.75 B.T.U. Answer. 336 ft. per sec.

19. Initial velocity of steam 200 ft. per sec., and heat drop 1.5 units. Calculate the exit velocity. Answer. 339 ft. per sec.

20. In a marine turbine 1 lb. of steam at an initial velocity of 200 ft. per sec. passes through the moving blades. Find the final velocity if the heat drop is 1.8 B.T.U. Answer. 360 ft. per sec.

Blade Velocities and Number of Rows.

RULE.—Velocity per sec.^{2} × Number of Rows = Constant.

Therefore, Constant ÷ Velocity^2 = Number of Rows,

and, $\sqrt{\text{Constant} \div \text{Number of Rows}}$ = Velocity.

NOTE.—The Constant referred to varies from 1,400,000 to 1,600,000 (see page 23).

EXAMPLE.—Determine the approximate total number of blade rows for a set of turbines (1 H.P. and 2 L.P.) if the surface velocity of the H.P. drum is

not to exceed 95 ft. and the L.P. drums 125 ft. per sec.; the Constant adopted is 1,500,000.

H.P.

Then, $95^2 \times \text{No.} = 1{,}500{,}000$.

Therefore, $1{,}500{,}000 \div 95^2 = 166$ rows, assuming total power in H.P. turbine.

Then, $160 \times \frac{1}{3} = 53$ rows, assuming $\frac{1}{3}$ power in H.P. turbine.

L.P.

$125^2 \times \text{No.} = 1{,}500{,}000$.

Therefore, $1{,}500{,}000 \div 125^2 = 96$ rows, assuming total power in L.P. turbines.

Then, $96 \times \frac{2}{3} = 64$ rows, assuming $\frac{2}{3}$ power in L.P. turbines.

1. Calculate the required total number of rows of ahead blades required for a Parsons marine turbine if the constant is 1,400,000 and the blade velocity 100 ft. per sec. Answer. 140 rows of blades.

2. Determine the total number of ahead blade rows required for the three rotors of a turbine if the blade velocity is to be 90 ft. per sec. and the constant allowed is 1,500,000. Answer. 185 rows of blades.

3. Find the suitable mean blade velocity if the number of rows is 192 and the constant 1,600,000. Answer. 91.2 ft. per sec.

4. What blade velocity per sec. will be necessary if the ahead turbine rotors contain in all 168 rows of blades and the given constant is 1,512,000? Answer. 94.8 ft. per sec.

5. The ahead H.P. and L.P. (2) turbine rotors each contain 48 rows of blades. Calculate the required mean blade velocity if the constant is 1,500,000. Answer. 102 ft. per sec.

6. Determine the number of blade rows required in each ahead rotor if the mean blade velocity is 120 ft. per sec. and the constant 1,600,000. Answer. 111 rows in all; 37 rows in each rotor.

7. Determine the constant used when the mean blade velocity is 105 ft. per sec. and the number of rows 144. Answer. 1,587,600 constant.

8. The number of ahead blade rows is 156 and the mean blade velocity 100 ft. per sec. Calculate the constant which has been employed. Answer. 1,560,000 constant.

9. Determine the number of blade rows required in the H.P. turbine and in each L.P. turbine, given that the H.P. rotor diameter is 42 in., the L.P. rotor diameter 60 in., the revolutions per min. 500, and the constant 1,600,000.

Answers. { H.P. turbine, 65 rows.
L.P. turbines (each), 63 rows.

10. Calculate the number of blade rows required in the H.P. and L.P. turbines of cruisers of the "Indomitable" and "Inflexible" class, given that the H.P. rotor is 92 in. diameter, the L.P. rotor 115 in. diameter, the revolutions (designed) 260 per min., and the constant adopted 1,300,000.

NOTE.—2 H.P. and 2 L.P. ahead turbines.

Answers. { H.P. turbines, 60 rows.
L.P. „ 38 „

Turbine Propeller Calculations.

RULE.— $\sqrt{\text{Effective thrust on shaft}} = C \times$ Diameter of propeller.

Therefore, $\dfrac{\sqrt{\text{Effective thrust}}}{C} =$ Diameter of propeller.

And, $\dfrac{\sqrt{\text{Effective thrust}}}{\text{Diameter}} = C.$

NOTE.—C = Constant found from suitable coefficient for blade areas ratio and blade pressure per square inch.

EXAMPLE.—Calculate the required diameter of each propeller of the "Lusitania," given that the effective thrust on each shaft is 121,460 lbs. and the Constant is 23.

Then, Diam. of propeller $= \dfrac{\sqrt{\text{Thrust lbs.}}}{C} = \dfrac{\sqrt{121460}}{23} = 15.1$ ft. diam.

1. Determine the diameter of turbine propeller suitable if the effective thrust on each shaft is 23,180 lbs. and the constant 29.
Answer. 5.24 ft. diameter.

2. Calculate the diameter of turbine propeller required when the thrust per shaft is 24,000 lbs. and the constant C 30. Answer. 5.16 ft. diameter.

3. Find the required diameter of propeller if the effective thrust is 60,000 lbs. on each shaft and the constant 24.63.
Answer. 9.94 ft. diameter.

4. Calculate the propeller diameter if the effective thrust in each shaft is found to be 44,000 lbs. and the constant 29. Answer. 7.23 ft. diameter.

5. What diameter of propeller should be employed when the thrust is 30,000 lbs. on each shaft and the constant is 30?
Answer. 5.77 ft. diameter.

6. Determine the effective thrust pounds if the propeller diameter is 5 ft. and the constant 31. Answer. 24,025 lbs. thrust on each shaft.

7. What is the effective thrust in lbs. if the propeller diameter is 14 ft. and the constant 21? Answer. 86,436 lbs. thrust on each shaft.

8. What diameter of propeller should be used when the effective thrust per shaft is 25,000 lbs. and the constant 28? Answer. 5.64 ft. diameter.

9. What diameter of propeller should be fitted to each of three shafts when the effective thrust is 91,476 lbs. and the constant is 21?
Answer. 14.4 ft. diameter.

10. What diameter of propeller should be used when the effective thrust is 188,051 lbs. and the constant 25, the steamer having 4 lengths of shafting and 4 propellers? Answer. 17.3 ft. diameter.

Pressure Drops.

RULE.—

$$\frac{\text{Absolute initial pressure.} - \text{Absolute final pressure}}{\text{Number of blade rows}} = \text{Mean drop per row.}$$

or, Drop per row × Number of rows + Final pressure = Initial pressure.

NOTE.—Take atmospheric pressure as 15 lbs. per square inch.

EXAMPLE.—Determine the mean pressure drop per blade row in the H.P. and L.P. turbines, given the following:—H.P. initial pressure, 145 lbs. gauge;

H.P. terminal pressure, 16 lbs. gauge; L.P. initial pressure, 15 lbs. gauge; L.P. initial pressure, 27 in. vacuum. Each turbine contains 60 rows of blades.

Then, $145 + 15 = 160$ lbs. absolute initial pressure.
And, $16 + 15 = 31$ „ terminal „

Therefore, Mean pressure drop in H.P. turbine $= \dfrac{160 - 31}{60} = 2.15$ lbs.

Again, $15 + 15 = 30$ lbs. absolute initial pressure.

And, $\left(\dfrac{30 \text{ in.} - 27 \text{ in.}}{2}\right) = 1.5$ „ terminal „

Therefore, Mean pressure drop in L.P. turbines $= \dfrac{30 - 1.5}{60} = .475$ lb.

NOTE.—Barometer height = 30 in. at ordinary atmospheric pressure.

1. Calculate the mean pressure drop per row if the H.P. turbine initial gauge pressure is 140 lbs. and the L.P. turbine initial gauge pressure is 20 lbs., the rotor containing 48 rows of blades. Answer. 2.5 lbs. drop per row.

2. The H.P. rotor consists of 4 expansions, each containing 16 rows of blades. Calculate the mean pressure drop per each expansion and per each row if the initial pressure is 150 lbs. gauge and the final pressure 22 lbs. gauge. Answer. 32 lbs. drop per expansion; 2 lbs. drop per row.

3. The L.P. turbine gauges show 20 lbs. pressure and the condenser vacuum gauge 27 in. Calculate the mean pressure drop per expansion and per row if the rotor consists of 8 expansions, each containing 8 rows of blades.
Answer. 4.18 lbs. drop per expansion; .523 lb. drop per row.

4. In running "full astern" the reverse turbine gauges show 80 lbs. pressure and the condenser gauges 27 in. vacuum. Calculate the pressure drop per expansion and per blade row if the L.P. rotors are made up of 8 expansions, each containing 7 rows of blades.
Answer. 11.68 lbs. drop per expansion; 1.66 lbs. drop per row.

5. In running "half speed ahead" the H.P. turbine gauge indicates 40 lbs., and the L.P. turbine gauges each 12 in. vacuum with a condenser vacuum of 28 in. Calculate the pressure drop per blade row if the H.P. and L.P. rotors each contain 56 rows of blades.

Answers. $\begin{cases} \text{.82 lb. drop per row in H.P. turbines.} \\ \text{.142 „ „ L.P. „} \end{cases}$

6. In running "slow ahead" the H.P. turbine gauge shows 10 lbs. pressure and the L.P. turbine gauges each 21 in. vacuum, the condenser vacuum being 28 in. Calculate the pressure drop per expansion and per blade row if the H.P. contains 4 expansions of 16 rows each, and the L.P.'s contain 8 expansions of 8 rows each.

Answers. $\begin{cases} \text{5.125 lbs. drop per expansion, H.P. turbines.} \\ \text{.32 lb. drop per row, H.P. turbines.} \\ \text{.437 lb. drop per expansion, L.P. turbines.} \\ \text{.0546 lb. drop per row, L.P. turbines.} \end{cases}$

7. Calculate the required initial H.P. pressure by gauge, if the L.P. initial pressure is 16 lbs. gauge and the mean pressure drop per row is to be 2 lbs. The H.P. rotor consists of 4 expansions, each containing 16 rows of blades.
Answer. 144 lbs. gauge pressure.

8. The L.P. turbines contain 8 expansions of 8 rows of blades per expansion. If the condenser vacuum is 27 in. and the mean pressure drop per row to be .46 lb., calculate the required L.P. initial pressure by gauge.

Answer. 15.94 lbs. gauge pressure.

9. The H.P. rotor contains 4 expansions of 12 blade rows in each, and the L.P. rotor 8 expansions of 6 blade rows in each. Calculate the required H.P. and L.P. turbine initial gauge pressures if the mean H.P. pressure drop per row requires to be 2.6 lbs. and the mean L.P. pressure drop per row .703 lb., the condenser vacuum being $27\frac{1}{2}$ in.

Answers. { L.P. initial pressure, 19.99 or 20 lbs. gauge.
H.P. „ 144.79 or 145 lbs. gauge. }

Heat Drop Calculations.

(Adiabatic Expansion.)

RULE 1.—Heat Drop $= H_1 \times f_1 - H_2 \times f_2 + T_1^\circ - T_2^\circ =$ B.T.U.

Where H_1 = Latent Heat before Expansion,
H_2 = Latent Heat after Expansion,
T_1° = Absolute Temperature before Expansion,
T_2° = Absolute Temperature after Expansion,
f_1 = Dryness Factor before Expansion,
f_2 = Dryness Factor after Expansion.

NOTE.—Absolute temperature = Fahrenheit + 461°.

„ See Table of Properties of Saturated Steam for Latent Heat and Temperatures Fahrenheit.

RULE 2.—

$$\frac{\text{Pounds steam per minute} \times \text{Effective heat drop} \times 778}{33000} = \text{Turbine horse-power.}$$

NOTE.—For L.P. turbines take *half* total weight of steam.

RULE 3.—Total pounds steam per hour ÷ Combined horse-power of all turbines = Steam consumption per horse-power hour.

RULE 4.—Steam consumption per hour ÷ Evaporation = Coal consumption per horse-power hour.

RULE 5.—Effective foot lbs. (or horse-power) ÷ Theoretical foot lbs. (or horse-power) = Efficiency.

NOTE.—The foregoing rules neglect gland and blade tip leakage, &c.

EXAMPLE.—The H.P. initial pressure is 140 lbs. by gauge, the terminal pressure 20 lbs. by gauge, the dryness factors before and after expansion .99 and .95, the steam consumption 90,000 lbs. per hour, and turbine over-all efficiency .6. Calculate (1) the heat drop per lb. of steam; (2) the effective foot lbs. given up per lb. steam; (3) the effective turbine horse-power developed in the H.P. turbine; and (4) the steam consumption per horse-power per hour, assuming equal horse-powers in each of the 3 turbines.

Then, 140 + 15 = 155 lbs. absolute initial pressure,
and, 155 lbs. = Temperature Fahr. 361.1, and Latent Heat 859.6.

Again, 361.1 + 461 = 822.1 absolute Temperature.

Then, 20 + 15 = 35 lbs. absolute final pressure,
and, 35 lbs. = 259.3 Temperature Fahr., and 931.6 Latent Heat.

Again, $259.3 + 461 = 720.3$ absolute Temperature.

Therefore,

$$\begin{aligned}\text{Heat drop} &= 859.6 \times .99 - 931.6 \times .95 + 822.1 - 720.3 \\ &= 851.004 - 885.020 + 822.1 - 720.3 \\ &(851.004 + 822.1 - 885.02 + 720.3) \\ &= 1673.104 - 1605.32 = 67.78 \text{ B.T.U. per lb.}\end{aligned}$$

Again, $67.78 \times 778 \times .65 = 34876.3$ Foot lbs. given up.

Then, $90000 \div 60 = 1500$ lbs. Flow per minute.

$$\text{Horse-power} = \frac{1500 \times 34876.3}{33000} = 1585$$

Steam consump. per horse-power per hour $= 90000 \div (1585 \times 3) = 18.9$ lbs.

1. Calculate the heat drop per lb. of steam occurring during one H.P. turbine expansion, if the pressure falls from 135 lbs. gauge to 125 lbs. gauge, the dryness factor being 1 before expansion and .996 after expansion.

Answer. 4.96 B.T.U. given up.

2. Calculate the heat drop per lb. of steam throughout one H.P. expansion if the pressure falls from 160 lbs. absolute to 130 lbs. absolute, the dryness factors being 1 and .98. Answer. 21.87 B.T.U. given up.

3. Calculate the total heat drop per lb. of steam throughout the H.P. turbine if the initial pressure by gauge is 135 lbs. and the final pressure by gauge is 12 lbs., the dryness factors being 1 and .9.

Answer. 127.42 B.T.U. given up.

4. Calculate the total heat drop per lb. of steam which occurs in the L.P. turbines if the initial pressure is 12 lbs. gauge and terminal pressure 27 in. vacuum, the dryness factor being .9 and .78.

Answer. 170.58 B.T.U. given up.

5. Calculate the heat drop per lb. of steam which takes place (1) in the H.P. turbine, and (2) in each L.P. turbine, assuming that the L.P. turbines each receive only half the weight of steam used by the H.P. turbine, given that the H.P. initial pressure is 140 lbs. gauge, the L.P. turbine initial pressure 16 lbs. gauge, and the condenser vacuum 28 in.; the dryness factors are assumed to be .95, .9, and .76.

Answers. (1) 82.49 B.T.U. in H.P. turbine.
(2) 100.263 „ each L.P. turbine.

6. Taking the efficiency as 56 per cent., calculate (1) the effective work done per lb. of steam in the previous question, and (2) the horse-power developed if 60,000 lbs. of steam are passing through the turbines per hour; also (3) the steam consumption per horse-power per hour.

Answers. (1) 123304.4108 ft. lbs.
(2) 3736.49 horse-power.
(3) 16.05 lbs. steam per hour per H.P.

7. Calculate the B.T.U. given up and transformed into kinetic energy per lb. of steam, also the combined horse-power of the turbines, for the following:—

H.P. turbine initial pressure, 150 lbs. gauge.
L.P. turbine exhaust pressure, 26 in. vacuum.
Dryness factors, .99 and .76 respectively.
Turbine efficiency, 54 per cent.
Steam consumption, 90,000 lbs. per hour.

Answers. 129283.52 ft. lbs. kinetic energy per lb. steam.
5876.5 horse-power.

8. Calculate the effective B.T.U. per lb. of steam, and the steam consumption per hour per horse-power for the following:—

H.P. turbines initial pressure, 145 lbs. gauge.
L.P. turbines exhaust pressure, 25 in. vacuum.
Dryness factors, .98 and .78.
Turbine efficiency, 56 per cent.

Also, Find coal consumption per horse-power per hour, assuming the evaporation to be 8.6.

Answers. { 153.508 B.T.U. per pound.
16.57 lbs. steam per hour per H.P.
1.92 lbs. coal per hour per H.P. }

9. Calculate the B.T.U. transformed into kinetic energy in each L.P. turbine per lb. of steam supplied to the H.P. turbine, if the L.P. turbine initial pressure is 20 lbs. gauge, and the vacuum at the exhaust end 26 in. Dryness factors .89 and .76, and efficiency 55 per cent. Also calculate the horse-power developed in the L.P. turbines, if the steam supplied to each is 43,500 lbs. per hour.

Answers. { 50.19 B.T.U. per each L.P. turbine.
1715.73 horse-power ,, }

10. During trials, gauges fitted to the L.P. turbine casings show the following pressures:—

L.P. turbines	1st	expansion	(initial),	18 lbs. gauge.
	2nd	,,	,,	11 ,,
	3rd	,,	,,	5 ,,
	4th	,,	,,	0 ,,
	5th	,,	,,	8 in. vacuum.
	6th	,,	,,	15 ,,
	7th	,,	,,	21 ,,
	8th	,,	,,	24 ,,
	,,	,,	(terminal),	26 ,,

Calculate the heat drops per lb. of steam at each expansion, and the theoretical ft. lbs. of energy developed throughout the L.P. turbines per lb. of steam supplied, assuming dryness factors of .81, .8, .79, .78, .77, .76, .75, .74, .73.

Answers.

15.18	B.T.U.	1st	expansion.
15.76	,,	2nd	,,
16.24	,,	3rd	,,
16.55	,,	4th	,,
17.97	,,	5th	,,
20.27	,,	6th	,,
17.75	,,	7th	,,
17.56	,,	8th	,,
137.28	B.T.U.	total heat drop.	
106803.84 ft. lbs.			

11. During the trials of a turbine steamer gauges are fitted to the casings at each expansion, and the pressures registered on the H.P. were as follows:—

H.P. turbine	1st expansion	(initial),	140 lbs.	(gauge).
	2nd "	"	100 "	"
	3rd "	"	65 "	"
	4th "	"	40 "	"
	" "	(terminal),	20 "	"

Assume the dryness factors to be .9, .88, .86, .84, and .82 respectively, and calculate the heat drops per lb. of steam which take place at each expansion, and the total heat drop throughout this turbine, neglecting friction and other losses.

Answers.
25.94 B.T.U. 1st expansion.
27.69 " 2nd "
27.91 " 3rd "
29.96 " 4th "

111.5 B.T.U. total heat drop.

12. During the trials of a turbine steamer gauges were fitted to the casings at each expansion, and the pressures registered on the H.P. turbine were as follows:—

1st expansion (initial),	140 lbs.		4th expansion (initial),	40 lbs.
2nd " "	100 "		" " (terminal),	20 "
3rd " "	65 "			

Assume the dryness factors to be 1, .98, .96, .945, and .92, and calculate the heat drops per lb. of steam which take place at each expansion, and the total heat drop throughout the H.P. turbine, neglecting frictional losses.

Answers.
24.31 B.T.U. 1st expansion.
25.85 " 2nd "
21.58 " 3rd "
32.56 " 4th "

104.30 B.T.U. total heat drop.

13. Running at "half-speed," gauges fitted on the H.P. turbine casing registered 35 lbs., 22 lbs., 10 lbs., and 0 lbs. at each expansion, and the H.P. terminal pressure 12 in. vacuum. Assume the dryness factors to be .9, .88, .869, .85, and .839, and calculate (1) the theoretical heat drop at each expansion, (2) the total heat drop, and (3) the horse-power developed in the H.P. turbine if the turbine efficiency is 70 per cent., and the steam consumption 70,000 lbs. per hour.

Answers.
25.286 B.T.U. 1st expansion.
18.8183 " 2nd "
28.810 " 3rd "
20.64 " 4th "

93.55 total heat drop.

1801.1 horse-power.

14. Each L.P. turbine uses 35,000 lbs. of steam per hour, and the pressures shown on the casing gauges at each expansion were 14, 16, 17, 19, 21, 23, 24, 26,

and 27 in. vacuum, the dryness factors being .836, .825, .812, .802, .792, .78, .765, .755, and .744. Calculate (1) the heat drops occurring at each expansion, (2) the total heat drops, and (3) the effective horse-power developed in each L.P. turbine, the turbine efficiency being 76 per cent.

Answers.

13.29	B.T.U.	1st	expansion.
14.302	,,	2nd	,,
13.03	,,	3rd	,,
13.72	,,	4th	,,
16.624	,,	5th	,,
17.89	,,	6th	,,
17.296	,,	7th	,,
16.17	,,	8th	,,

122.322 total heat drop.

1278.49 horse-power.

15. Running "full speed ahead," the H.P. turbine gauges showed 150 lbs., 110 lbs., 75 lbs., 45 lbs., and 23 lbs. Assuming the dryness factors to be 1, .98, .97, .94, and .92, calculate the heat drops at each expansion and the total heat drops throughout the H.P. turbine.

Answers.

23.93	B.T.U.	1st	expansion.
23.98	,,	2nd	,,
28.065	,,	3rd	,,
28.076	,,	4th	,,

104.051 total heat drop.

16. Referring to the previous question, if the L.P. turbine expansion gauges showed pressures of 22 lbs., 14 lbs., 8 lbs., 3 lbs., 2 in., 8 in., 14 in., 21 in., and a terminal pressure equal to 28 in. vacuum, and the dryness factors are assumed to be .92, .905, .89, .875, .86, .84, .81, .78, and .76, calculate the heat drop at each expansion and the total heat drop in each L.P. turbine; also, the horse-power being 6,500, the turbine efficiency 56 per cent., and the evaporation 8.6 lbs. of water per lb. of coal, calculate (1) steam used per hour, (2) lbs. of steam per horse-power per hour, (3) coal consumption per horse-power per hour, and (4) tons burned per day of 24 hours.

Answers.

18.999	B.T.U.	1st	expansion.
18.892	,,	2nd	,,
19.2775	,,	3rd	,,
19.3395	,,	4th	,,
24.08	,,	5th	,,
35.651	,,	6th	,
40.765	,,	7th	,,
45.845	,,	8th	,,

222.8490 B.T.U. in each L.P. turbine.

(1) 90,360 lbs. of steam per hour.
(2) 13.9 lbs. of steam per H.P. per hour.
(3) 1.61 lbs. of coal per H.P. per hour.
(4) 112.1 tons of coal per day.

17. Running "full astern" with the two L.P. reverse turbines, the gauge pressures were 80 lbs. at the initial end, and 26 in. vacuum at the exhaust end. Calculate (1) the total heat drop in each L.P. turbine per lb. of steam supplied assuming dryness factors of .97 and .76, (2) the horse-power obtained in each turbine if the total steam consumption is 1,340 lbs. per min., and (3) the lbs. of steam per horse-power per hour if the efficiency of the turbines is 58 per cent.

Answers.
(1) 227.418 B.T.U. total heat drop.
(2) 2,541 horse-power in each L.P.
(3) 15.8 lbs. of steam per horse-power per hour.

18. The H.P. initial pressure is 145 lbs. gauge and the L.P. terminal pressure is 27 in. vacuum. Calculate (1) the mean turbine efficiency if the horse-power developed is 8,000, and total steam consumption 2,000 lbs. per minute. Also (2) determine the steam consumption per horse-power per hour assuming dryness factors of .99 and .76.

Answers.
(1) .54 mean turbine efficiency.
(2) 15 lbs. steam per horse-power per hour.

19. The shaft horse-power in a turbine channel steamer as measured by the "Denny-Johnson" torsion meter worked out as 6,500 and the coal consumption was 5.2 tons coal per hour. Calculate the turbine efficiency if the H.P. pressure is 140 lbs. gauge and the L.P. turbine terminal pressure equal to 27 in. vacuum. Assume dryness factors of .98 and .75. Evaporation 8.5 lbs. of water per lb. of coal. Answer. Turbine efficiency .53.

20. H.P. turbine initial pressure 140 lbs. gauge, H.P. turbine terminal pressure 21 lbs. gauge, L.P. turbine initial pressure 20 lbs. gauge, and the condenser vacuum 28 in. Assuming dryness factors to be .99, .91, .9, .75, the consumption 43.2 tons for 6 hours' steaming, evaporation 8.5 lbs. of water per lb. of coal, and the horse-power by "Denny-Johnson" torsion meter 10,000, calculate (1) the total theoretical heat drop per lb. of steam, and (2) ft. lbs. of energy given out, also (3) the consumption of steam, and (4) coal per horse-power hour, and (5) mean turbine efficiency.

Answers.
(1) 317.914 B.T.U. total theoretical heat drop.
(2) 247337.09 ft. lbs.
(3) 13.7 lbs. steam per H.P. hour.
(4) 1.61 lbs. coal per H.P. hour.
(5) 58 per cent. mean turbine efficiency.

21. Calculate the effective B.T.U. given up in each turbine per lb. of steam supplied to the H.P. turbine, the horse-power developed in each, and the lbs. steam per horse-power hour, given the following:—

H.P. initial pressure 140 lbs. gauge.
L.P. " " 20 " "
" terminal " 1½ " absolute.

Assuming adiabatic expansion the dryness factors are 1, .93, and .78. The turbines consume 112,000 lbs. of steam per hour, and the over-all efficiency is 60 per cent.

Answers.
57 B.T.U. in H.P. turbine.
61.16 B.T.U. in each L.P. turbine.
2508.3 horse-power in H.P. turbine.
2691.6 horse-power in each L.P. turbine.
14.1 lbs. steam per horse-power per hour.

Areas between Blades and Blade Heights.

RULE 1.—

$$\frac{\text{Horse-power} \times \text{Lbs. steam per hour} \times \text{Volume}}{60 \times 60} = \text{Cub. ft. flow per sec.}$$

RULE 2.— $$\frac{\text{Constant}}{\sqrt{N}} = \text{Initial velocity per sec.}$$

NOTE.—Constant = 2700 to 3000.

" N = No. of blade rows by rule given (page 55).

RULE 3.—

$$\frac{\text{Cub. ft. flow per sec.}}{\text{Initial velocity}} = \text{Required clear area between blades.}$$

RULE 4.—

$$\sqrt{\frac{\text{Area of rotor drum circle} + \text{Clear exit area} \times \text{Annulus factor}}{.7854}} = \text{Diameter across blade tips.}$$

RULE 5.—

$$\frac{\text{Diameter across tips} - \text{Rotor drum diameter}}{2} = \text{Required blade heights.}$$

NOTE.—The annulus factor varies from about 3 for H.P. turbine blades to 2 or less for L.P. blades, and depends on the blade thickness and exit angle, which angle varies from about 20 degs. in the 1st H.P. expansion to about 65 degs. in the last L.P. expansion.

" Work all in feet and decimals (3 or 4 places).

RULE 6.—Blade heights of other expansions. Blade height = 1st expansion blade height × Blade ratio.

NOTE.—The blade height ratio is usually 1.41 or 1.41, but in special designs may only be 1.34.

RULE 7.—Blade heights of 1st L.P. Expansions.

$$\frac{\text{Last H.P. blade height} \times \text{Blade ratio}}{\text{Drum ratio} \times \text{Drum ratio} \times \text{No. L.P. turbines}} = \text{1st L.P. blade height.}$$

For other L.P. blades apply Rule 6.

EXAMPLE.—The shaft horse-power is 6,500, the steam consumption 13 lbs. per horse-power hour, the revolutions 400, and the initial pressure 160 lbs. (gauge). Determine (1) the blade heights at the 1st H.P. expansion, if the annulus factor is 3, and the drum diameter is to be 3 ft. 9 in.; also calculate (2) the inside diameter of casing at 1st expansion, and (3) the ratio V_t, V_s. Velocity constant = 2700, and N = 150.

$$\text{Cubic feet flow per second} = \frac{6500 \times 13 \times 2.56}{60 \times 60} = 60.08 \text{ cub. ft.}$$

$$\text{Initial steam velocity} = \frac{2700}{\sqrt{150}} = 220 \text{ ft. per sec.}$$

$$\text{Area between blades} = 60.08 \div 220 = .273 \text{ sq. ft.}$$

$$\text{Diameter across blades} = \sqrt{\frac{3.75^2 \times .7854 + .273 \times 3}{.7854}} = 3.88 \text{ ft.}$$

$$\text{Blade height} = \frac{(3.886 - 3.75) \times 12}{2} = .816, \text{ say } \tfrac{7}{8} \text{ in.}$$

$$\text{Inside diameter of casing} = 45 + (.875 \times 2) = 46\tfrac{3}{4} \text{ in.}$$

$$\text{Ratio } V_t, V_s = \left(\frac{3.75 \times 3.1416 \times 400}{60}\right) \div 220 = .35.$$

1. The turbine horse-power is 10,800, revolutions 500 per minute, and the steam consumption 13 lbs. per horse-power hour; the H.P. turbine initial steam pressure is 140 lbs. by gauge, and the corresponding steam volume per lb. 2.878 cub. ft.; the annular area factor is 3, and the rotor drum diameter is to be 4 ft. Calculate (1) the required blade heights, and (2) the inner diameter of the turbine casing at the 1st expansion (neglecting blade tip clearance). Velocity constant = 2700, and N = 144 rows.

Answers. (1) $1\frac{3}{8}$ in. blade height at 1st expansion.
(2) $50\frac{3}{4}$ in. diameter of casing (inside).

2. Calculate the required blade heights at the 2nd, 3rd, and 4th expansions of this turbine, the blade height ratio being 1.4.

Answers.

2nd expansion blades,	1.92	in., say	2	in.
3rd "	2.80	"	$2\frac{3}{4}$	"
4th "	3.85	"	$3\frac{7}{8}$	"

3. If the L.P. rotor drum of the same turbine set is 68 in. diameter, calculate the blade heights of the 1st, 2nd, 3rd, 4th, 5th, and 6th expansions, the 7th and 8th expansions being "wing" blades.

Answers.

1st expansion blades,	1.38	in., say	$1\frac{3}{8}$	in.
2nd "	1.92	"	2	"
3rd "	2.80	"	$2\frac{3}{4}$	"
4th "	3.85	"	$3\frac{7}{8}$	"
5th "	5.42	"	$5\frac{1}{2}$	"
6th "	7.70	"	$7\frac{3}{4}$	"
7th "	"wing,"		$7\frac{3}{4}$	"
8th "	"		$7\frac{3}{4}$	"

4. The turbine horse-power is 5,000, the revolutions 450 per minute, the steam consumption 14 lbs. per horse-power hour, the H.P. turbine initial pressure 133 lbs. by gauge (volume 2.99 cub. ft. per lb.), the annular factor is 3, and the H.P. rotor drum diameter is 3 ft. 6 in. Determine the required blade heights of the 1st H.P. expansion if the velocity constant is 2,700 and N = 158 rows. Answer. $\frac{7}{8}$ in. blades at 1st expansion.

5. Calculate the required blade heights of the 2nd, 3rd, and 4th expansions of the H.P. turbine, the blade height ratio being 1.41.

Answers.

2nd expansion blades,	1.22	in., say	$1\frac{1}{4}$	in.
3rd "			$1\frac{3}{4}$	"
4th "	2.45	"	$2\frac{1}{2}$	"

6. Calculate the required blade heights of the L.P. expansions, the drum being 5 ft. diameter; also, the 7th and 8th expansions being fitted with "wing" blades. Calculate the required annulus factor, assuming the steam velocity at these expansions being respectively 350 ft. and 380 ft. per sec., and the actual volumes 90 cub. ft. and 140 cub. ft. per lb.

Answers.

1st expansion blades,	.86	in., say	$\frac{7}{8}$	in.
2nd "	1.22	"	$1\frac{1}{4}$	"
3rd "			$1\frac{3}{4}$	"
4th "	2.45	"	$2\frac{1}{2}$	"
5th "			$3\frac{1}{2}$.	"
6th "	4.9	"	5	"
7th "			5	"
8th "			5	"

Annulus factor for 7th expansion, 2.8.
" " 8th " 1.95.

7. Determine the blade heights at each of the H.P. expansions if the horse-power is 8,000, allowing 13.5 lbs. of steam per horse-power hour. The rotor drum is 3 ft. 3 in. diameter, and N rows 156. Assume a constant of 2,700 for an initial pressure of 145 lbs. (gauge), and allow normal blades with exit area $\frac{1}{3}$ annulus area, ratio of blade heights to be 1.4.

Answers.	1st expansion blades,	1.29 in.,	say	$1\frac{1}{4}$	in.
	2nd ,,			$1\frac{3}{4}$	,,
	3rd ,,	2.45	,,	$2\frac{1}{2}$	,,
	4th ,,			$3\frac{1}{2}$	,,

8. Determine the blade heights at each of the L.P. expansions, 8 in number, referring to question No. 7, the L.P. drum being 4 ft. 7 in. diameter, and the blade height ratio as before. Allow equal blade heights at the 6th, 7th, and 8th expansions.

Answers.	1st expansion blades,	1.23 in.,	say	$1\frac{1}{4}$	in.
	2nd ,,	1.75	,,	$1\frac{3}{4}$	,,
	3rd ,,	2.45	,,	$2\frac{1}{2}$	,,
	4th ,,	3.5	,,	$3\frac{1}{2}$	,,
	5th ,,	4.9	,,	5	,,
	6th ,,	7.0	,,	7	,,
	7th ,,	"wing"		7	in.
	8th ,,	,,		7	,,

9. Calculate, by the rules given, the required blade heights for the H.P. turbine (4 expansions) of a steamer of 10,000 horse-power, the steam consumption being estimated at 13.2 lbs. per horse-power hour, the rotor drum 3 ft. 9 in. diameter, the initial pressure 145 lbs., the number of blade rows N 144, the steam initial velocity constant 2,700, and the blade height ratio 1.4 (normal blades).

Answers.	1st expansion blades,	1.32 in.,	say	$1\frac{3}{8}$	in.
	2nd ,,	1.92	,,	2	,,
	3rd ,,	2.8	,,	$2\frac{7}{8}$	,,
	4th ,,			4	,,

10. Calculate the blade heights for the L.P. turbines (8 expansions) of the foregoing, allowing the 6th, 7th, and 8th expansions to be of equal blade height, but the 6th and 7th expansions of increasing exit area; the L.P. rotor drums are 5 ft. 4 in. diameter.

Answers.	1st expansion blades,	1.39 in.,	say	$1\frac{3}{8}$	in.
	2nd ,,	1.92	,,	2	,,
	3rd ,,	2.8	,,	$2\frac{7}{8}$	,,
	4th ,,			4	,,
	5th ,,	5.6	,,	$5\frac{1}{2}$	,,
	6th ,,	7.7	,,	$7\frac{3}{4}$	,,
	7th ,,			$7\frac{3}{4}$	,,
	8th ,,			$7\frac{3}{4}$	,,

11. Determine the required blade heights for each of the four H.P. turbine expansions, the horse-power being 14,000, the steam consumption per horse-power hour estimated at 13.4 lbs., the initial H.P. pressure 135 lbs., the

velocity constant 2,800, the rotor drum 4 ft. 6 in., the blade height ratio 1.42, and N number of rows 140 (normal blades).

Answers.	1st expansion blades,	1.56 in.,	say	$1\frac{1}{2}$ in.
	2nd "	2.13 "		$2\frac{1}{4}$ "
	3rd "	3.19 "		$3\frac{1}{4}$ "
	4th "	4.61 "		$4\frac{5}{8}$ "

12. Repeat for each L.P. turbine, the drums being 6 ft. 6 in. diameter the 7th and 8th expansions fitted with "wing" blades of more than normal exit area, but of same height as those of the 6th expansion.

Answers.	1st expansion blades,	1.58, in.,	say	$1\frac{5}{8}$ in.
	2nd "	2.30 "		$2\frac{3}{8}$ "
	3rd "	3.37 "		$3\frac{3}{8}$ "
	4th "	4.79 "		$4\frac{3}{4}$ "
	5th "	6.74 "		$6\frac{3}{4}$ "
	6th "	9.58 "		$9\frac{1}{2}$ "
	7th "			$9\frac{1}{2}$ "
	8th "			$9\frac{1}{2}$ "

13. Determine the blade heights required for the H.P. turbine of four expansions of a coastal destroyer, if the turbine horse-power is 5,500, the estimated consumption 14 lbs. per horse-power hour, the initial pressure 180 lbs., the velocity constant 3,000, and blade rows N 121. The rotor drum is 3 ft. 8 in. diameter, and the blade height ratio 1.4.

Answers.	1st expansion blade heights,	.54 in.,	say	$\frac{1}{2}$ in.
	2nd " "	.7 "		$\frac{3}{4}$ "
	3rd " "	1.05 "		1 "
	4th " "	1.4 "		$1\frac{1}{2}$ "

14. Also work out the required L.P. turbine blade heights (eight expansions), the rotor drum being 5 ft. 6 in. diameter. The last two expansions are to be of the same height as the 6th expansion.

Answers.	1st expansion blade heights,	.46 in.,	say	$\frac{1}{2}$ in.
	2nd " "	.70 "		$\frac{3}{4}$ "
	3rd " "	1.05 "		1 "
	4th " "	1.4 "		$1\frac{1}{2}$ "
	5th " "	2.1 "		$2\frac{1}{8}$ "
	6th " "	2.97 "		3 "
	7th " "			3 "
	8th " "			3 "

15. Work out the necessary H.P. turbine blade heights for a cruiser of 22,000 horse-power, two H.P. and two L.P. turbines being fitted for ahead running. The H.P. turbines are to have six expansions, the blade height ratio being 1.4, the rotor drum diameter is 5 ft. 6 in., the estimated steam consumption 13.8 lbs. per horse-power hour, the initial pressure 185 lbs., the velocity constant 3,000, and number of blade rows N 144.

Answers.	1st expansion blade heights,	.78 in.,	say	$\frac{7}{8}$ in.
	2nd " "	1.22 "		$1\frac{1}{4}$ "
	3rd " "			$1\frac{3}{4}$ "
	4th " "	2.45 "		$2\frac{1}{2}$ "
	5th " "			$3\frac{1}{2}$ "
	6th " "	4.9 "		5 "

16. Proceed in the same way to determine the required blade heights of the six L.P. turbine expansions, the rotor drum being 7 ft. 5 in. The 5th and 6th expansions are fitted with "wing" blades.

Answers.					
	1st expansion	blade heights,	3.85	in., say	$3\frac{7}{8}$ in.
	2nd	,, ,,	5.42	,,	$5\frac{1}{2}$,,
	3rd	,, ,,	7.7	,,	$7\frac{3}{4}$,,
	4th	,, ,,	10.8	,,	11 ,,
	5th	,, ,,		,,	11 ,,
	6th	,, ,,		,,	11 ,,

17. Determine the required height of blades for the six expansions of the L.P. turbine of combined reciprocating and turbine machinery, the total horse-power being 8,300, and allowing $11\frac{1}{2}$ lbs. of steam per horse-power per hour. The turbine initial pressure is 7 lbs. absolute and actual volume 46 cub. ft. per lb., the rotor drum is 7 ft. 6 in. diameter, and the blade height ratio 1.33; assume an initial steam velocity of 600 ft. per second, and normal blades for the first five expansions, the 6th expansion being of "wing" type.

Answers.					
	1st expansion	blades,			3 in.
	2nd	,,	3.99	in. say	4 ,,
	3rd	,,	5.32	,,	$5\frac{1}{2}$,,
	4th	,,	7.3	,,	$7\frac{1}{2}$,,
	5th	,,	9.97	,,	10 ,,
	6th	,,			10 ,,

18. Determine the required blade heights for the five expansions of the turbine for a combined reciprocating and turbine engined steamer, the horse-power being 3,000, and the estimated steam consumption 13 lbs. per horse-power per hour. The turbine initial pressure is 8 lbs. absolute, and actual volume 41 cub. ft. per lb.; the rotor drum is to be 6 ft. in diameter, and the blade height ratio 1.4. Allow normal blades for the first four expansions, and assume steam velocity at 1st expansion 400 ft. per second.

Answers.					
	1st expansion	blades,	2.04	in., say	$2\frac{1}{8}$ in.
	2nd	,,	2.97	,,	3 ,,
	3rd	,,	4.2	,,	$4\frac{1}{4}$,,
	4th	,,	5.95	,,	6 ,,
	5th	,,			6 ,,

Turbine Shaft Horse-Power (Solid Shafts).

Measured by Denny-Johnson Torsion Meter.

RULE.— $\dfrac{1.53 \times r \times d^4 \times R}{C \times L} = \text{Horse-power.}$

Where r = Reading of Torsion Meter in inches (arc of torque).
,, d = Diameter of Shaft in inches.
,, R = Revolutions per minute.
,, C = Inductor Constant (radius of indication).
,, L = Length of Shaft between Inductor Wheels in feet.

NOTE.—Recent trials with reciprocating engines bring out the torsion meter horse-power as equal to about 90 per cent. of the actual I.H.P.
,, The approximate equivalent I.H.P. can therefore be determined as follows:—
Horse-power by torsion meter $\div .9$ = Equivalent I.H.P.
,, The majority of the examples which follow are taken from actual practice.

EXAMPLE.—Calculate the shaft or transmitted horse-power of the H.P.

turbine if the length between inductor wheels is 40 ft., the radius Constant $12\frac{1}{2}$ in., and the reading .5 of an inch torque. The revolutions are 450 and the diameter of shafting 8 in. Also express the equivalent I.H.P.

Then, $$\text{Horse-power} = \frac{1.53 \times r \times d^4 \times R}{C \times L}$$

$$= \frac{1.53 \times .5 \times 8^4 \times 450}{12.5 \times 40} = 2820 \text{ horse-power.}$$

And, $2820 \div .9 = 3133$ I.H.P.

1. Calculate the shaft horse-power of a turbine steamer as measured by torsion meter, the length between the indicator wheels being 36 ft., the inductor radius constant 12.5 in., and the reading of the recording dials shows .68 of an inch torque for all three shafts. The revolutions are: H P. 600, Starboard L.P. 650, and Port L.P. 654. The shafts are each $6\frac{3}{4}$ in. diameter. Also determine the total equivalent I.H.P.

Answers.

H.P.	Turbine	2879	horse-power.
S.L.P.	„	3119	„
P.L.P.	„	3138	„
	Total	9136	horse-power.
	And	10151	I.H.P.

2. The inductor wheels of the torsion meter are fixed to each shaft at a distance apart of 33 ft. The inductor radius constant is 12.5 in., and the reading of the dials indicates a torsional arc of .8 in. Calculate the shaft horse-power of each turbine if the H.P. revolutions are 690, S.L.P. 830, and P.L.P. 825; the diameter of each shaft being $4\frac{3}{4}$ in. Also calculate the total approximate I.H.P.

Answers.

H.P.	Turbine	1042	horse-power.
S.L.P.	„	1253	„
P.L.P.	„	1246	„
	Total	3541	horse-power.
		3934.4	I.H.P.

3. Calculate the turbine shaft horse-power for the following: L = 25 ft., C = 12.5, $r = .35$ for all shafts, R = 490 H.P., 580 S.L.P., and 582 P.L.P., $d = 6$ in. Also find the combined I.H.P. (approximate).

Answers.

H.P.	Turbine	1088	horse-power.
S.L.P.	„	1288	„
P.L.P.	„	1292	„
	Total	3668	horse-power.
		4075.5	I.H.P.

4. Determine the shaft horse-power by torsion meter if the shafts are 10 in. in diameter, and the revolutions 300 for each turbine. Distance L is

60 ft., inductor constant 12.5 in., and mean reading of dials indicate .6 of an inch torsional arc. Determine the total approximate I.H.P.

Answers.	H.P.	Turbine	3672	horse-power.
	S.L.P.	„	3672	„
	P.L.P.	„	3672	„
		Total	11016	horse-power.
			12240	I.H.P.

5. Calculate the total shaft horse-power of a turbine steamer if the revolutions per minute are 625 in H.P., 630 in S.L.P., and 632 in P.L.P. The length between inductor wheels is 40 ft., and the reading of torsion meter dials shows .358 of an inch torsion. Shaft 8 in. diameter and inductor constant 12.5 in. Also find the total equivalent I.H.P. (approximate).

Answers.	H.P.	Turbine	2804	horse-power.
	S.L.P.	„	2826	„
	P.L.P.	„	2835	„
		Total	8465	horse-power.
			9405.5	I.H.P.

6. Calculate the shaft horse-power of the "Victorian," given that the inductor constant is 12.5 in., the mean torsional reading .3 of an inch, and the length between inductor wheels 40 ft. The mean revolutions are 230 per min. for all shafts, and the tunnel shafting 11 in. diameter. Find also the total I.H.P. of the turbines.

Answers.	H.P.	Turbine	3091.3	horse-power.
	S.L.P.	„	3091.3	„
	P.L.P.	„	3091.3	„
		Total	9273.9	horse-power.
			10304.3	I.H.P.

7. Calculate the turbine shaft horse-power of the "Carmania," given that the length between inductor wheels is 100 ft., the inductor constant 18.5 in., and the mean dial reading 1 in. The tunnel shafting is 14½ in. diameter, and the average revolutions 190 per min. Find also the combined I.H.P. equivalent.

Answers.	H.P.	Turbine	6946	horse-power.
	S.L.P.	„	6946	„
	P.L.P.	„	6946	„
		Total	20838	horse-power.
			23153	I.H.P.

8. Calculate the turbine shaft horse-power of the "Lusitania," given that the inductor radius constant is 18.5 in., the length between wheels 80 ft., the mean revolutions of each shaft 160 per min., and the mean torsion meter

reading .438 of an inch. The four lengths of shafting are each 22 in. in diameter. Also calculate the total equivalent I.H.P. (approximate).

Answers.

S.H.P.	Turbine	16971	horse-power.
S.L.P.	„	16971	„
P.H.P.	„	16971	„
P.L.P.	„	16971	„
	Total	67884	horse-power.
		75426	I.H.P.

Horse-Power Developed in Blade Rows.

RULE 1.—
$$\frac{(V^2_{S3} - V^2_{S2}) \times \text{Blade efficiency} \times \text{Lbs. steam per minute}}{64.4 \times 33000}$$
= Horse-power per row of blades.

and, Horse-power per row × Number of rows per expansion
= Horse-power developed in one "expansion."

Where V_{S3} = Steam Exit Velocity per second.
„ V_{S2} = Steam Admission Velocity per second.
„ 64.4 = (32.2 × 2 Gravity's Acceleration per second per second.)
„ 33000 = Foot lbs. per minute per horse-power.

RULE 2.—
$$\left(\frac{\text{Blade mean diam. in ft.} \times 3.1416 \times \text{Revs.}}{60}\right) \div V_{S3} = \text{Ratio } V_t \text{ to } V_{s3}.$$

EXAMPLE.—Calculate (1) the horse-power developed in one blade row, (2) in one H.P. expansion of 10 rows, and (3) the total effective heat drop occurring in one expansion, given the following:—Steam initial velocity 160 ft. per sec., exit velocity 250 ft. per sec., blade efficiency 85 per cent., and steam flow 2,400 lbs. per minute. Also express the ratio V_t, V_s at this expansion if the mean blade diameter is 4 ft. and the revolutions are 480 per min.

Then (1)
$$\text{Horse-power per blade row} = \frac{(250^2 - 160^2) \times 2400 \times .85}{64.4 \times 33000} = 35.4 \text{ horse-power.}$$

And (2) Horse-power per expansion = 35.4 × 10 = 354 horse-power.
And (3) 354 × 33000 ÷ 778 = 15015 B.T.U.

1. Calculate (1) the horse-power developed at one row of blades of the 1st expansion of an H.P. turbine, given that the steam flow is 22.5 lbs. per sec., and the blade efficiency .847. The steam admission velocity is 217 ft. per sec., and the exit velocity 322 ft. per sec. Also calculate (2) the horse-power developed in the 1st expansion, which contains 8 rows of similar blades, and (3) the effective heat drop in B.T.U. If the mean blade diameter is 3 ft. $8\frac{1}{2}$ in. and the revolutions 650 per minute, also (4) express the ratio of V_t to V_{s3} at this expansion.

Answers.
(1) 30.45 horse-power per blade row.
(2) 243.6 horse-power per expansion.
(3) 10332.6 B.T.U. heat drop.
(4) .39 ratio V_t to V_{s3}.

2. Calculate (1) the horse-power given out at one row of blades of the 6th expansion of the same H.P. turbine, given that the initial velocity of the steam is now 245 ft. per sec., the exit velocity 360 ft. per sec., and blade efficiency .83. Also (2) find the horse-power of this expansion, which contains 8 rows of blades, and (3) the effective heat drop. Express (4) the ratio of V_t to V_{s3} if the mean blade diameter is 3 ft. $9\frac{1}{4}$ in.

Answers.
(1) 36.68 horse-power per blade row.
(2) 293.44 horse-power per expansion.
(3) 12446.6 B.T.U. heat drop.
(4) .356 ratio V_t to V_{s3}.

3. Referring to question No. 1, calculate (1) the horse-power developed at one row of blades of the L.P. 1st expansion, given that each L.P. turbine receives only half the steam supplied to the H.P. turbine. The steam initial velocity is 350 ft. per sec., the exit velocity 487 ft. per sec., and blade efficiency .82. Also (2) find the horse-power given out by one expansion of 8 blade rows, and (3) the corresponding heat drop in B.T.U. Finally (4) express the ratio of V_t to V_{s3} if the mean diameter across the blades is 4 ft. $5\frac{1}{4}$ in.

Answers.
(1) 29.86 horse-power per blade row.
(2) 238.88 horse-power per expansion.
(3) 10132.44 B.T.U. heat drop.
(4) .310 ratio V_t to V_{s3}.

4. Find (1) the horse-power developed between one row of blades of the 4th L.P. expansion, given that the steam flow is as before stated, and the blade efficiency .76. The steam admission velocity is 397 ft. per sec., and the exit velocity 541 ft. per sec. Also (2) find the horse-power of this expansion of 8 blade rows, and (3) the effective heat drop. Finally (4) determine the ratio V_t to V_{s3} if the mean blade diameter is 4 ft. $7\frac{1}{2}$ in.

Answers.
(1) 32.6 horse-power per blade row.
(2) 260.8 horse-power per expansion.
(3) 11062.2 B.T.U. heat drop.
(4) .29 ratio V_t to V_{s3}.

5. Repeat the foregoing for the 5th L.P. expansion, if $V_{s2} = 475$ ft., $V_{s3} =$ 600 ft., blade efficiency .74, and mean blade diameter = 4 ft. 8 in. Eight rows of blades.

Answers.
(1) 31.58 horse-power per blade row.
(2) 252.64 horse-power per expansion.
(3) 107160.9 B.T.U. heat drop.
(4) .264 ratio V_t to V_{s3}.

6. Repeat the foregoing for the 6th L.P. expansion, if $V_{s2} = 646$ ft., $V_{s3} =$ 750 ft., blade efficiency .715, and mean blade diameter = 4 ft. 9 in. Eight rows of blades.

Answers.
(1) 32.97 horse-power per blade row.
(2) 263.76 horse-power per expansion.
(3) 11187.7 B.T.U. heat drop.
(4) .215 ratio V_t to V_{s3}.

Steam Condensed in Turbines.

Rule, lbs. steam × (1 − dryness factor) = water produced by nature of expansion.

1. Determine the amount of water drained off from the L.P. turbines by the air pumps per hour when the steam flow per hour is 112,000 lbs. and the dryness factor of the steam at the last L.P. expansion is .85.

Answer, 16,800 lbs.

2. The horse-power is 6,000 and the steam consumption is $13\frac{1}{2}$ lbs. per horse-power hour; determine the pounds of water condensed in the turbines per hour if the pressure at the last L.P. expansion is 3 lbs. absolute.

NOTE.—Take dryness factor from table of "Actual Steam Volumes," page 41. Answer, 12,312 lbs.

3. Calculate the amount of water drained off per hour from each L.P. turbine of the "Indomitable," given that the horse-power at 27 knots was 45,000 and steam consumption 13 lbs. per horse-power per hour. Assume dryness factor at last L.P. expansion of .835. Answer, 21.54 tons.

4. The horse-power is 10,000 and the steam consumption 14 lbs. per horse-power per hour; calculate the weight of steam condensed in the H.P. turbine per minute if the pressure at the last H.P. expansion is 20 lbs. (gauge) and the dryness factor .932, assuming the initial steam to be dry.

Answer, 158.6 lbs. per minute.

5. Refer to the previous question, and estimate the water condensed in the two L.P. turbines per minute if the dryness factor at the last L.P. expansions is .85. Answer, 350 lbs. per minute.

6. Determine the total water drained off by the "wet" air pumps from the L.P. turbines of the "Lusitania" per hour if the horse-power is 68,000, and allowing 13.7 lbs. of steam per horse-power per hour. Dryness factor, .848. Answer, 63.2 tons.

SOLUTIONS TO PROBLEMS.

Steam and Blade Speeds.

1. $V_t = 300 \times .4 = 120$ ft. per sec. Answer.

2. $V_s = 90 \div .45 = 200$ ft. per sec. Answer.

3. $V_t = 250 \times .48 = 120$ ft. per sec. Answer.

4. Ratio $V_t\ V_s = 150 \div 350 = .42$. Ratio. Answer.

Diameter of Rotors, &c.

1. Diameter $= \dfrac{9000}{3.1416 \times 400} = 7.16$ ft. Answer.

2. Diameter $= \dfrac{6000}{3.1416 \times 550} = 3.47$ ft. Answer.

3. Diameter $= \dfrac{5000}{3.1416 \times 400} = 3.97$ ft. Answer.

4. Revolutions $= \dfrac{8000}{6 \times 3.1416} = 424$. Answer.

5. Revolutions $= \dfrac{9000}{7 \times 3.1416} = 409$. Answer.

6. Revolutions $= \dfrac{100 \times 60}{3.1416 \times 5.5} = 347$. Answer.

7. Revolutions $= \dfrac{90 \times 60}{3.1416 \times 4} = 429$. Answer.

8. Revolutions $= \dfrac{160 \times 60}{3.1416 \times 14} = 218$. Answer.

Steam Velocities and Heat Drops.

1. $\dfrac{1 \times 600^2}{64.4} = 5590$ ft. lbs. Answer.

2. $\dfrac{1 \times 300^2}{64.4} = 1397$ ft. lbs.
$1.397 \div 778 = 1.79$ B.T.U. Answer.

3. $\dfrac{1 \times 350^2}{64.4} = 1902$ ft. lbs.
$1902 \div 778 = 2.44$ B.T.U. Answer.

4. $\dfrac{1 \times (\frac{15000}{60})^2}{64.4} = 970$ ft. lbs.
$970 \div 778 = 1.24$ B.T.U. Answer.

5. $\dfrac{(300^2 - 200^2) \times 1}{64.4} = 776$ ft. lbs.
$776 \div 778 = .99$ B.T.U. Answer.

6. $\dfrac{(400^2 - 300^2) \times 1}{64.4} = 1086$.
$1086 \div 778 = 1.39$ B.T.U. Answer.

7. $\dfrac{(300^2 - 220^2) \times 1}{64.4} = 645$.
$\therefore\ 645 \div 778 = .82$ B.T.U. Answer.

8. $\sqrt{\dfrac{64.4 \times 2000}{1}} = 358$ ft. per sec. Answer.

9. $\sqrt{\dfrac{64.4 \times 1000}{1}} = 253$ ft. per sec. Answer.

10. $\sqrt{\dfrac{64.4 \times 10000}{20}} = 179$ ft. per sec. Answer.

11 $2 \times 778 = 1556$ ft. lbs.,

and $\sqrt{\dfrac{64.4 \times 1556}{1}} = 316$ ft. per sec. Answer.

12. $2.5 \times 778 = 1945$ ft. lbs.,

and $\sqrt{\dfrac{64.4 \times 1945}{1}} = 353$ ft. per sec. Answer.

13. $\sqrt{64.4 \times 1500 + 200^2} = 369$ ft. per sec. Answer.

14. $2 \times 778 = 1556$ ft. lbs.,

and $\sqrt{64.4 \times 1556 + 300^2} = 436$ ft. per sec. Answer.

15. $1.5 \times 778 = 1167$ ft. lbs.,

and $\sqrt{64.4 \times 1167 + 260^2} = 377$ ft. per sec. Answer.

16. $3 \times 778 = 2334$ ft. lbs.,

and $\sqrt{64.4 \times 1167 + (\frac{6000}{60})^2} = 400$ ft. per sec. Answer.

17. 1200

1196

4 B.T.U. given up,

and $4 \times 778 = 3112$ ft. lbs. of energy given up.

Then, $\sqrt{\dfrac{3112 \times 64.4 + 300^2}{1}} = 538$ ft. per sec. Answer.

18. $778 \times 1.75 = 1361.5$ ft. lbs.,

$\sqrt{\dfrac{1361.5 \times 64.4 + 160^2}{1}} = 336$ ft. per sec. Answer.

19. $778 \times 1.5 = 1167$ ft. lbs.,

and $\sqrt{\dfrac{1167 \times 64.4 + 200^2}{1}} = 339$ ft. per sec. Answer.

20. $778 \times 1.8 = 1400.4$ ft. lbs.,

and $\sqrt{\dfrac{1400.4 \times 64.4 + 200^2}{1}} = 360$ ft. per sec. Answer.

Blade Velocities and Number of Blade Rows.

1. $\dfrac{1400000}{100^2} = 140$ rows. Answer.

2. $\frac{1500000}{100^2} = 150$ rows. Answer.

3. $\sqrt{\frac{1600000}{192}} = 91.2$ ft. per sec. Answer.

4. $\sqrt{\frac{1500000}{168}} = 94.4$ ft. per sec. Answer.

5. $\sqrt{\frac{1500000}{48 \times 3}} = 102$ ft. per sec. Answer.

6. $\sqrt{\frac{1600000}{120^2}} = 111$ rows in all, and $\frac{111}{3} = 37$ rows each. Answer.

7. $144 \times 90^2 = 1166400$ constant. Answer.

8. $156 \times 100^2 = 1560000$ constant. Answer.

Blade Velocities and Number of Rows.

1. $1400000 \div 100^2 = 140$ rows. Answer.

2. $1500000 \div 90^2 = 185$ rows. Answer.

3. $\sqrt{1600000 \div 192} = 91.2$ ft. Answer.

4. $\sqrt{1512000 \div 168} = 94.8$ ft. Answer.

5. $\sqrt{1500000 \div (48 \times 3)} = 102$ ft. Answer.

6. $\sqrt{1600000 \div 120^2} = 111$ rows in all,
and $111 \div 3 = 37$ rows in each turbine. Answer.

7. $105^2 \times 144 = 1587600$ constant. Answer.

8. $100^2 \times 156 = 1560000$ constant. Answer.

9. $\frac{42 \times 3.1416 \times 500}{12 \text{ in.} \times 60} = 91.6$ ft. H.P. velocity,

$\frac{60 \times 3.1416 \times 500}{12 \text{ in.} \times 60} = 130.9$ ft. L.P. velocity.

Then, $1600000 \div 91.6^2 = 196$ rows for total power,
and $196 \times \frac{1}{3} = 65$ rows for $\frac{1}{3}$ power.
Again, $1600000 \div 130.9^2 = 94$ rows for total power,
and $94 \times \frac{2}{3} = 63$ (nearly) rows for $\frac{2}{3}$ power.

10. $\frac{92 \times 3.1416 \times 260}{12 \times 60} = 104$ ft. H.P. velocity,

$\frac{115 \times 3.1416 \times 260}{12 \times 60} = 130$ ft. L.P. velocity.

Then, $1500000 \div 104^2 = 120$ rows for total power,
and $120 \div 2 = 60$ rows for $\frac{1}{2}$ power.
Again, $1500000 \div 130^2 = 76$ rows for total power,
and $76 \div 2 = 38$ rows for $\frac{1}{2}$ power.

Turbine Propeller Calculations.

1. $\frac{\sqrt{23180}}{29} = 5.24$ ft. diameter. Answer.

2. $\frac{\sqrt{24000}}{30} = 5.16$ ft. diameter. Answer.

3. $\frac{\sqrt{60000}}{24.63} = 9.94$ ft. diameter. Answer.

4. $\frac{\sqrt{44000}}{29} = 7.23$ ft. diameter. Answer.

5. $\frac{\sqrt{30000}}{30} = 5.77$ ft. diameter. Answer.

6. $(5 \times 31)^2 = 24025$ lbs. thrust. Answer.

7. $(14 \times 21)^2 = 86436$ lbs. thrust. Answer.

8. $\frac{\sqrt{25000}}{28} = 5.64$ ft. diameter. Answer.

9. $\frac{\sqrt{91476}}{21} = 14.4$ ft. diameter. Answer.

10. $\frac{\sqrt{188051}}{25} = 17.3$ ft. diameter. Answer.

Pressure Drops.

1. $\frac{(140+15)-(20+15)}{48} = 2.5$ lbs. per row. Answer.

2. $\frac{(150+15)-(22+15)}{16 \times 4} = 2$ lbs. per row. Answer.

And $\frac{(150+15)-(22+15)}{4} = 32$ lbs. per expansion. Answer.

3. 27 in. vacuum = 1.5 lbs. absolute.

$\therefore \frac{(20+15)-1.5}{8} = 4.18$ lbs. per expansion. Answer.

And $\frac{(20+15)-1.5}{8 \times 8} = .523$ lb. per row. Answer.

4. NOTE.—27 in. vacuum = 1.5 lbs. absolute.

$\therefore \frac{(80+15)-1.5}{7 \times 8} = 1.66$ lbs. per row. Answer.

And $\frac{(80+15)-1.5}{8} = 11.68$ lbs. per expansion. Answer.

5. NOTE.—12 in. vacuum = 9 lbs. absolute.
28 in. vacuum = 1 lb. absolute.

$\therefore \dfrac{(40+15)-9}{56} = .82$ lb. per row in H.P. Answer.

And $\dfrac{9-1}{56} = .142$ lb. per row in L.P. Answer.

6. NOTE.—21 in. vacuum = 4.5 lbs. absolute.
28 in. vacuum = 1 lb. absolute.

Then, $\dfrac{(10+15)-4.5}{4} = 5.125$ lbs. per expansion in H.P. Answer.

And $\dfrac{(10+15)-4.5}{4 \times 16} = .32$ lb. per row in H.P. Answer.

Then, $\dfrac{4.5-1}{8} = .437$ lb. per expansion in L.P. Answer.

And $\dfrac{4.5-1}{8 \times 8} = .0546$ per row in L.P. Answer.

7. $4 \times 16 \times 2 + (16+15) = 159$ lbs. absolute.
$\therefore$ $159 - 15 = 144$ lbs. gauge. Answer.

8. NOTE.—27 in. vacuum = 1.5 lbs. absolute.
Then, $8 \times 8 \times .46 + 1.5 = 30.94$ lbs. absolute.
$\therefore$ $30.94 - 15 = 15.94$ lbs. gauge. Answer.

9. NOTE.—$27\frac{1}{2}$ in. vacuum = 1.25 lbs. absolute.
Then, $8 \times 6 \times .703 + 1.25 = 34.99$ lbs. absolute.
$\therefore$ $34.99 - 15 = 19.99$ or 20 lbs. gauge on L.P. Answer.
Again, $4 \times 12 \times 2.6 + 34.99 = 159.79$ lbs. absolute.
$\therefore$ $159.79 - 15 = 144.79$ or 145 lbs. gauge on H.P. Answer.

Heat Drop Calculations.

1. $135 + 15 = 150$ lbs., and $125 + 15 = 140$ lbs.
$358.3 + 461 = 819.3^\circ$; $352.9 + 461 = 813.9$.
$\therefore$ $861.5 \times 1 - 865.4 \times .996 + 819.3 - 813.9 = 4.96$ B.T.U. Answer.

NOTE.—861.5 = latent heat at 150 lbs.
865.4 = latent heat at 140 lbs.

2. $160 + 15 = 175$ lbs; $130 + 15 = 145$ lbs.
$370.8 + 461 = 831.8^\circ$; $355.6 + 461 = 816.6^\circ$.
$\therefore$ $852.9 \times 1 - 863.5 \times .98 + 831.8 - 816.6 = 21.87$ B.T.U. Answer.

NOTE.—852.9 = latent heat at 175 lbs.
863.5 = latent heat at 145 lbs.

3. $135 + 15 = 150$ lbs.; $12 + 15 = 27$ lbs.
$358.3 + 461 = 819.3^\circ$; $244.4 + 461 = 705.4^\circ$.
$\therefore$ $861.5 \times 1 - 942.2 \times .9 + 819.3^\circ - 705.4^\circ = 127.42$ B.T.U. Answer.

NOTE.—861.5 = latent heat at 150 lbs.
942.2 = latent heat at 27 lbs.

4. $12+15=27$ lbs.; 27 in. vacuum $=1.5$ lbs. absolute.
$244.4+461=705.4$; $115.9+461=576.9$.
$\therefore 942.2\times.9-1033.2\times.78+705.4-576.9=170.58$ B.T.U. Answer.

5. $140+15=155$ lbs.; $16+15=31$ lbs.
$361+461=822.1°$; $252+461=713.2$.
$\therefore 859.6\times.95-936.7\times.9+822.1-713.2=82.49$ B.T.U. in H.P.
Answer.

NOTE.—28 in. vacuum $=1$ lb. absolute pressure.
$102.1+461=563.1°$.

Then, $936.71\times.9-1042.9\times.76+713.2-563.1=\frac{200.52}{2}$
$=100.263$ B.T.U. in L.P. Answer.

6. Total B.T.U.'s $=82.49$ for H.P., and 200.526 for two L.P.'s
$=283.016$ B.T.U.
$\therefore 283.016\times778\times.56=123304.4108$ ft. lbs.

And $\frac{123304.4108\times60000}{33000\times60}=3736.49$ horse-power.

Then, $\frac{60000}{3736.49}=16.05$ lbs. steam per hour per horse-power. Answer

7. $150+15=165$ lbs.; 26 in. vacuum $=2$ lbs. absolute.
$366+461=827$; $126.3+461=587.3$.
$\therefore 856.2\times.99-1025.8\times.76+827-587.3=307.73$ B.T.U.
And $307.73\times778\times.54=129283.5276$ ft. lbs. kinetic energy. Answer.

Again, $\frac{129283.5276\times90000}{33000\times60}=5876.52$ horse-power. Answer.

8. $145+15=160$ lbs.; 25 in. vacuum $=2\frac{1}{2}$ lbs.
$363.6+461=824.6$; $134.6+461=595.6$.
$\therefore 857.8\times.98-1019.9\times.78+824.6-595.6=153.508$ B.T.U.'s.
And $33000\times60\div153.508\times778=16.57$ lbs. steam per hour per H.P.,
and $16.57\div8.6=1.92$ lbs. coal per hour per H.P. Answer.

9. $20+15=35$ lbs.; 26 in. vacuum $=2$ lbs.
$259+461=720.3$; $126.3+461=587.3$.
$\therefore 931.6\times.89-1025.8\times.76+720.3-587.3\times.55=50.19$ B.T.U.

Then, $\frac{50.19\times778\times43500\times2}{33000\times60}=1715.73$ horse-power in L.P. turbines.
Answer.

10. 1st Expansion—
$18+15=33$ lbs. $=255.9+461=716.9°$; $934°=$ latent heat at 33 lbs.
$11+15=26$ lbs. $=242.3+461=703.3°$; $943.7=$ latent heat at 26 lbs.
$\therefore 934\times.81-943.7\times.8+716.9-703.3=15.18$ B.T.U. Answer.

2nd Expansion—
$11+15=26$ lbs. $=242.3+461=703.3°$.
$5+15=20$ lbs. $=228+461=689°$; 953.8 latent heat at 12 lbs.
$\therefore 943.7\times.8-953.8\times.79+703.3-689=15.76$ B.T.U. Answer.

3rd Expansion—

0 lbs. gauge = 15 lbs. absolute = 213.1 + 461 = 674.1.

$\therefore$ 953.8 × .79 − 964.3 × .78 + 698 − 674.1 = 16.24 B.T.U. Answer.

4th Expansion—

8 in. vacuum = 11 lbs. absolute = 197.8 + 461 = 658.8.

$\therefore$ 964.3 × .78 − 975.2 × .77 + 674.1 − 658.8 = 16.55 B.T.U. Answer.

5th Expansion—

15 in. vacuum = 7½ lbs. = 180 + 461 = 641°.

$\therefore$ 975.2 × .77 − 987.8 × .76 + 658.8 − 641 = 17.97 B.T.U. Answer.

6th Expansion—

21 in. vacuum = 4½ lbs. = 157.9 + 461 = 618.9.

$\therefore$ 987.8 × .76 − 1003.4 × .75 + 641 − 618.9 = 20.27 B.T.U. Answer.

7th Expansion—

24 in. vacuum = 3 lbs. = 141.6 + 461 = 602.6.

$\therefore$ 1003.4 × .75 − 1015 × .74 + 618.9 − 602.6 = 17.75 B.T.U. Answer.

8th Expansion—

26 in. vacuum = 2 lbs. = 126.3 + 4.61 = 587.3.

$\therefore$ 1015 × .74 − 1025.8 × .73 + 602.6 − 587.3 = 17.56 B.T.U. Answer.

Then,

15.18
15.76
16.24
16.55
17.97
20.27
17.75
17.56

137.28 B.T.U. Total heat drop. Answer.

$\therefore$ 137.28 × 778 = 106803.84 ft. lbs. Answer.

11. 1st Expansion—

140 + 15 = 155 lbs. = 361.1 + 461 = 822.1; 859.6° = latent heat at 155 lbs.

100 + 15 = 115 lbs. = 338 + 461 = 799; 875.9° = latent heat at 115 lbs.

$\therefore$ 859.6 × .9 − 875.9 × .88 + 822.1 − 799 = 25.94 B.T.U. Answer.

2nd Expansion—

65 + 15 = 80 lbs. = 312 + 461 = 773°.

$\therefore$ 875.9 × .88 − 894.3 × .86 × 799 − 773 = 27.69 B.T.U. Answer.

3rd Expansion—

40 + 15 = 55 lbs. = 287.1 + 461 = 748.1.

$\therefore$ 894.3 × .86 − 912 × .84 + 773 − 748.1 = 27.91 B.T.U. Answer.

4th Expansion—

$20+15=35$ lbs. $=259.3+461=720.3$.

$\therefore\ 912\times.84-931.6\times.82+748.1-720.3=29.96$ B.T.U. Answer.

Then
25.94
27.69
27.91
29.96

111.50 B.T.U. total expansion. Answer.

12. 1st Expansion—

$140+15=155$ lbs. $=361.1+461=822.1°$.

$100+15=115$ lbs. $=338+461=799$.

$\therefore\ 859.6\times1-875.9\times.98+822.1-799=24.31$ B.T.U. Answer.

2nd Expansion—

$65+15=80$ lbs. $=312+461=773°$.

$\therefore\ 875.9\times.98-894.3\times.96+799-773=25.85$ B.T.U. Answer.

3rd Expansion—

$40+15=55$ lbs. $=287.1+461=748.1$.

$\therefore\ 894.3\times.96-912\times.945+773-748.1=21.58$ B.T.U. Answer.

4th Expansion—

$20+15=.35$ lb. $=259.3+461=720.3$.

$\therefore\ 912\times.945-931.6\times.92+748.1-720.3=32.56$ B.T.U. Answer.

Then,
24.31
25.85
21.58
32.56

Total expansion 104.30 B.T.U. Answer.

13. 1st Expansion—

$35+15=50$ lbs. $=281°+461=742$.

$22+15=37$ lbs. $=262.6+461=723.6$.

$\therefore\ 916.3\times.9-929.3\times.88+742-723.6=25.286$ B.T.U. Answer.

2nd Expansion—

$22+15=37$ lbs. $=262.6+461=723.6$.

$10+15=25$ lbs. $=240.1+461=701.1$.

$\therefore\ 929.3\times.88-945.3\times.869+723.6-701.1=18.8183$ B.T.U. Answer.

3rd Expansion—

$10+15=25$ lbs. $=240.1+461=701.1$.

0 lbs. $+15=15$ lbs. $=213.1+461=674.1$.

$\therefore\ 945.3\times.869-964.3\times.85+701.1-674.1=28.810$ B.T.U. Answer.

4th Expansion—

$0+15=15$ lbs. $=213.1+461=674.1$.

12 in. vacuum $=9$ lbs. $=188.3+461=649.3$.

$\therefore 964.3 \times .85 - 981.9 \times .839 + 674.1 - 649.3 = 20.64$ B.T.U. Answer.

$$\begin{array}{r} 25.286 \\ 18.8183 \\ 28.810 \\ 20.64 \\ \hline 93.55 \end{array}$$ B.T.U. total heat drop.

Then, $\dfrac{93.55 \times 778 \times .70 \times 70000}{33000 \times 60} = 1801.1$ horse-power. Answer.

14. 1st Expansion—

14 in. vacuum $= 8$ lbs. $= 182.9 + 461 = 643.9°$; latent heat $= 985.7°$.
16 in. vacuum $= 7$ lbs. $= 176.9 + 461 = 637.9°$; latent heat $= 990°$.

$\therefore 985.7 \times .836 - 990 \times .825 + 643.9 - 637.9 = 13.29$ B.T.U. Answer.

2nd Expansion—

16 in. vacuum $= 7$ lbs. $= 176.9 + 461 = 637.9°$.
17 in. vacuum $= 6.5$ lbs. $= 173.6 + 461 = 634.6°$.

$\therefore 990 \times .825 - 992.3 \times .812 + 637.9 - 634.6 = 14.302$ B.T.U. Answer.

3rd Expansion—

17 in. vacuum $= 6.5$ lbs. $= 173.6 + 461 = 634.6°$.
19 in. vacuum $= 5.5$ lbs. $= 166.4 + 461 = 627.4°$.

$\therefore 992.3 \times .812 - 997.4 \times .802 + 634.6 - 627.4 = 13.03$ B.T.U. Answer.

4th Expansion—

19 in. vacuum $= 5.5$ lbs. $= 166.4 + 461 = 627.4°$.
21 in. vacuum $= 4.5$ lbs. $= 157.9 + 461 = 618.9°$.

$\therefore 997.4 \times .802 - 1003.4 \times .792 + 627.4 - 618.9 = 13.72$ B.T.U. Answer.

5th Expansion—

21 in. vacuum $= 4.5$ lbs. $= 157.9 + 461 = 618.9°$.
23 in. vacuum $= 3.5$ lbs. $= 147.7 + 461 = 608.7°$.

$\therefore 1003.4 \times .792 - 1010.6 \times .78 + 618.9 - 608.7 = 16.624$ B.T.U. Answer.

6th Expansion—

23 in. vacuum $= 3.5$ lbs. $= 147.7 + 461 = 608.7°$.
24 in. vacuum $= 3$ lbs. $= 141.6 + 461 = 602.6°$.

$\therefore 1010.6 \times .78 - 1015 \times .765 + 608.7 - 602.6 = 17.89$ B.T.U. Answer.

7th Expansion—

24 in. vacuum $= 3$ lbs. $= 141.6 + 461 = 602.6°$.
26 in. vacuum $= 2$ lbs. $= 126.3 + 461 = 587.3°$.

$\therefore 1015 \times .765 - 1025.8 \times .755 + 602.6 - 587.3 = 17.29$ B.T.U. Answer.

8th Expansion—

26 in. vacuum $= 2$ lbs. $= 126.3 + 461 = 587.3°$.
27 in. vacuum $= 1.5$ lbs. $= 115.9 + 461 = 576.9°$.

$\therefore\ 1025.8 \times .755 - 1033.2 \times .744 + 587.3 - 576.9 = 16.17$ B.T.U. Answer.

Then,
13.29
14.302
13.03
13.72
16.624
17.89
17.29
16.17

Total 122.322 B.T.U.

$$\therefore\ \frac{122.322 \times 778 \times .76 \times 35000}{33000 \times 60} = 1278.49 \text{ horse-power.} \quad \text{Answer.}$$

15. 1st Expansion—

$150 + 15 = 165$ lbs. $= 366 + 461 = 827^\circ$.
$110 + 15 = 125$ lbs. $= 344.2 + 461 = 805.2$.
$\therefore\ 856.2 \times 1 - 871.5 \times .98 + 827 - 805.2 = 23.93$ B.T.U. Answer.

2nd Expansion—

$110 + 15 = 125$ lbs. $= 344.2 + 461 = 805.2$.
$75 + 15 = 90$ lbs. $= 320.2 + 461 = 781.2$.
$\therefore\ 871.5 \times .98 - 880.5 \times .97 + 805.2 - 781.2 = 23.98$ B.T.U. Answer.

3rd Expansion—

$75 + 15 = 90$ lbs. $= 320.2 + 461 = 781.2$.
$45 + 15 = 60$ lbs. $= 292.7 + 461 = 753.7$.
$\therefore\ 880.5 \times .97 - 908 \times .94 + 781.2 - 753.7 = 28.065$. Answer.

4th Expansion—

$45 + 15 = 60$ lbs. $= 292.7 + 461 = 753.7$.
$23 + 15 = 38$ lbs. $= 264.2 + 461 = 725.2$.
$\therefore\ 908 \times .94 - 928.2 \times .92 + 753.7 - 725.2 = 28.076$. Answer.

Then,
23.93
23.985
28.065
28.076

104.056 B.T.U. total drop. Answer.

16. 1st Expansion—

$22 + 15 = 37$ lbs. $= 262.6 + 461 = 723.6^\circ$; latent heat $= 929.3^\circ$.
$14 + 15 = 29$ lbs. $= 248.4 + 461 = 709.4^\circ$; latent heat $= 939.4^\circ$.
$\therefore\ 929.3 \times .92 - 939.4 \times .905 + 723.6 - 709.4 = 18.999$ B.T.U. Answer.

2nd Expansion—

$14 + 15 = 29$ lbs. $= 248.4 + 461 = 709.4$.
$8 + 15 = 23$ lbs. $= 235.5 + 461 = 696.5$.
$\therefore\ 939.4 \times .905 - 948.5 \times .89 + 709.4 - 696.5 = 18.892$ B.T.U. Answer.

3rd Expansion—

$8 + 15 = 23$ lbs. $= 235.5 + 461 = 696.5$.

$3 + 15 = 18$ lbs. $= 222.4 + 461 = 683.4$.

$\therefore\ 948.5 \times .89 - 957.7 \times .875 + 696.5 - 683.4 = 19.2775$ B.T.U. Answer.

4th Expansion—

$3 + 15 = 18$ lbs. $= 222.4 + 461 = 683.4$.

2 in. vacuum $= 14$ lbs. $= 209.6 + 461 = 670.6$.

$\therefore\ 957.7 \times .875 - 966.8 \times .86 + 683.4 - 670.6 = 19.3395$ B.T.U. Answer.

5th Expansion—

2 in. vacuum $= 14$ lbs. $= 209.6 + 461 = 670.6$.

8 in. vacuum $= 11$ lbs. $= 194.8 + 461 = 658.8$.

$\therefore\ 966.8 \times .86 - 975.2 \times .84 + 670.6 - 658.8 = 24.08$ B.T.U. Answer.

6th Expansion—

8 in. vacuum $= 11$ lbs. $= 194.8 + 461 = 658.8$.

14 in. vacuum $= 8$ lbs. $= 182.9 + 461 = 643.9$.

$\therefore\ 975.2 \times .84 - 985.7 \times .81 + 658.8 - 643.9 = 35.651$ B.T.U. Answer.

7th Expansion—

14 in. vacuum $= 8$ lbs. $= 182.9 + 461 = 643.9$.

21 in. vacuum $= 4\frac{1}{2}$ lbs. $= 157.9 + 461 = 618.9$.

$\therefore\ 985.7 \times .81 - 1003.4 \times .78 + 643.9 - 618.9 = 40.765$ B.T.U. Answer.

8th Expansion—

21 in. vacuum $= 4\frac{1}{2}$ lbs. $= 157.9 + 461 = 618.9$.

28 in. vacuum $= 1$ lb. $= 102.1 + 461 = 563.1$.

$\therefore\ 1003.4 \times .78 - 1042.9 \times .76 + 618.9 - 563.1 = 45.848$ B.T.U. Answer.

$$\begin{array}{r} 18.999 \\ 18.892 \\ 19.2775 \\ 19.3395 \\ 24.08 \\ 35.651 \\ 40.765 \\ 45.848 \\ \hline \end{array}$$

Total 222.8490 B.T.U. Answer.

Then, 222.849 L.P. $+ 104.056$ H.P. $= 326.905$ B.T.U. total,

and $326.905 \times 778 \times .56 \times \text{lbs.} = 33000 \times 6500 \times 60$.

$$\therefore\ \frac{33000 \times 6500 \times 60}{326.905 \times 778 \times .56} = 90360 \text{ lbs. steam per hour.}$$

Again, $90360 \div 6500 = 13.9$ lbs. steam per horse-power per hour.

$13.9 \div 8.6 = 1.61$ lbs. coal per horse-power per hour.

And $1.61 \times 6500 \times 24 \div 2240 = 112.1$ tons coal per day. Answer.

17. $80 + 15 = 95$ lbs. $= 324.1 + 461 = 785.1$; latent heat $= 885.8°$.

26 in. vacuum $= 2$ lbs. $= 126.3 + 461 = 587.3$; latent heat $= 1025.8°$.

$\therefore\ 885.8 \times .97 - 1025.8 \times .76 + 785.1 - 587.3 = 277.418$ B.T.U. Ans.

Then, each L.P. turbine gets $\frac{1340}{2} = 670$ lbs. steam.

$\therefore \frac{277.418 \times 778 \times .58 \times 670}{33000} = \left\{ \begin{array}{l} 2541 \text{ horse-power in each L.P.} \\ \text{turbine. Answer.} \end{array} \right.$

Then, $670 \times 60 \div 2541 = 15.8$ lbs. of steam per horse-power per hour. Ans.

18. $145 + 15 = 160$ lbs. $= 363.6 + 461 = 824.6$; latent heat $= 857.8°$.
27 in. vacuum $= 1\frac{1}{2}$ lbs. $= 115.9 + 461 = 576.9$; latent heat $= 1033.2°$.
$\therefore 857.8 \times .99 - 1033.2 \times .76 + 824.6 - 576.9 = 311.690$ B.T.U.
Then, $8000 \times 33000 \div 2000 \times 311.69 \times 778 = .54$ efficiency. Answer.
And $2000 \times 60 \div 8000 = 15$ lbs. steam per horse-power per hour. Ans.

19. $140 + 15 = 155$ lbs. $= 361.1 + 461 = 822.1$.
27 in. vacuum $= 1\frac{1}{2}$ lbs. $= 115.9 + 461 = 576.9$.
$\therefore 859.6 \times .98 - 1033.2 \times .75 + 822.1 - 576.9 = 312.708$ B.T.U.
Then, $\frac{5.2 \times 2240}{60} = 194.13$ lbs. of coal per minute,
and as 8.5 lbs. of water are evaporated per lb. of coal,
$\therefore 194.13 \times 8.5 = 1650.1$ lbs. of steam used per minute.
Then, $33000 \times 6500 \div 312.708 \times 778 \times 1650.1 = .53$ efficiency. Answer.

20. $140 + 15 = 155$ lbs. $= 361.1 + 461 = 822.1$.
$21 + 15 = 36$ lbs. $= 260.9 + 461 = 721.9$.
$\therefore 859.6 \times .99 - 930.5 \times .91 + 822.1 - 721.9 = \left\{ \begin{array}{l} 104.449 \text{ B.T.U. in} \\ \text{H.P. turbine.} \end{array} \right.$
Again, $20 + 15 = 35$ lbs. $= 259.3 + 461 = 720.3$.
28 in. vacuum $= 1$ lb. $= 102.1 + 461 = 563.1$.
$\therefore 931.6 \times .9 - 1042.9 \times .75 + 720.3 - 563.1 = \left\{ \begin{array}{l} 213.465 \text{ B.T.U. in} \\ \text{L.P. turbines.} \end{array} \right.$
Then, total heat drop $= 104.449 + 213.465 = 317.914$ B.T.U. Answer.
$317.914 \times 778 = 247337.09$ ft. lbs. of energy given up. Answer.
Then, $43.2 \times 2240 \div 6 \times 60 = 268.8$ lbs. coal per minute.
$\therefore 268.8 \times 8.5 = 2284.8$ lbs. steam per minute.
And $2284.8 \times 60 \div 10000 = 13.7$ lbs. steam per H.P. per hour. Answer.
$13.7 \div 8.5 = 1.61$ lbs. of coal per horse-power per hour. Answer.
$33000 \times 10000 \div 317.914 \times 778 \times 2284.8 = \left\{ \begin{array}{l} .58 \text{ efficiency} = 58\,\%. \\ \text{Answer.} \end{array} \right.$

21. $140 + 15 = 155$ lbs. $= 361 + 461 = 822.1°$; latent heat $= 859.6$.
$20 + 15 = 35$ lbs. $= 259.3 + 461 = 720.3$; latent heat $= 931.6$.
$\therefore 859.6 \times 1 - 931.6 \times .93 + 822.1 - 720.3 = 95.01$.
And $95.01 \times .60 = 57$ B.T.U. in H.P. turbine. Answer.
$20 + 15 = 35$ lbs. $= 259.3 + 461 = 720.3°$; latent heat $= 931.6°$.
$1\frac{1}{2}$ lbs. $= 115.9 + 461 = 576.9°$; latent heat $= 1033.2°$
$\therefore 931.6 \times .93 - 1033.2 \times .78 + 720.3 - 576.9 = 203.892$.
$\therefore \frac{203.892 \times .60}{2} = 61.16$ B.T.U. in each L.P. turbine. Answer.

Again, $112000 \div 60 = 1866.6$ lbs. steam per minute.

$$\therefore \frac{1866.6 \times 57 \times 778}{33000} = 2508.3 \text{ horse-power in H.P. turbine. Answer.}$$

Then, $\frac{61.16 \times 778 \times 1866.3}{33000} = \begin{cases} 2691.6 \text{ horse-power in each L.P.} \\ \text{turbine. Answer.} \end{cases}$

Total horse-power $= 2508 + (2691.6 \times 2) = 7891.5$ horse-power.

$\therefore$ $112000 \div 7891.5 = 14.1$ lbs. steam per horse-power per hour. Ans.

Blade Areas and Blade Heights.

1. Steam flow per second $= \frac{10800 \times 13 \times 2.87}{60 \times 60} = 111.93$ cub. ft.

Steam initial velocity $= \frac{2700}{\sqrt{144}} = 225$ ft. per sec.

Area between blades $= 111.93 \div 225 = .497$ sq. ft.

Then, diameter across blades $= \sqrt{\frac{4^2 \times .7854 + .497 \times 3}{.7854}} = 4.23$ ft.

Blade height $= \frac{(4.23 - 4) \times 12 \text{ in.}}{2} = 1.38$ in., say $1\frac{3}{8}$ in. Ans.

Diam. of casing (at 1st expansion) $= 48$ in. $+ (1\frac{3}{8}$ in. $\times 2) = 50\frac{3}{4}$ in.

2. 2nd expansion blades $= 1.375 \times 1.4 = 1.925$ in., say 2 in. }
3rd " $= 2 \times 1.4 = 2.80$ " $2\frac{3}{4}$ " } Ans.
4th " $= 2.75 \times 1.4 = 3.85$ " $3\frac{7}{8}$ " }

3. Drum ratio $= 68$ in. $\div 48$ in. $= 1.4$.

$$\text{1st expansion blade heights} \left\{ \begin{array}{lll} = \frac{3.875 \times 1.4}{1.4 \times 1.4 \times 2} & = 1.38 \text{ in., say} & 1\frac{3}{8} \text{ in.} \\ = 1.375 \times 1.4 & = 1.92 \quad " & 2 \quad " \\ = 2 \times 1.4 & = 2.80 \quad " & 2\frac{3}{4} \quad " \\ = 2.75 \times 1.4 & = 3.85 \quad " & 3\frac{7}{8} \quad " \\ = 3.875 \times 1.4 & = 5.42 \quad " & 5\frac{1}{2} \quad " \\ = 5.5 \times 1.4 & = 7.70 \quad " & 7\frac{3}{4} \quad " \\ & 7.70 \quad " & 7\frac{3}{4} \quad " \\ & 7.70 \quad " & 7\frac{3}{4} \quad " \end{array} \right\} \text{Ans.}$$

4. Steam flow per second $= \frac{5000 \times 14 \times 2.99}{60 \times 60} = 58.14$ cub. ft.

Steam initial velocity $= \frac{2700}{\sqrt{158}} = \frac{2700}{12.56} = 215$ ft.

Area between blades $= 58.14 \div 215 = .2704$ sq. ft.

Diameter across blades $= \sqrt{\frac{(3.5^2 \times .2854) + (.2704 \times 3)}{.2854}} = 3.644$ ft.

Blade height $= \frac{(3.644 - 3.5) \times 12 \text{ in.}}{2} = .86$ in., say $\frac{7}{8}$ in. Ans.

5. 2nd expansion blades $= .875 \times 1.4 = 1.22$ in., say $1\frac{1}{4}$ in.
3rd " $= 1.25 \times 1.4 = 1.75$ " $1\frac{3}{4}$ "
4th " $= 1.75 \times 1.4 = 2.45$ " $2\frac{1}{2}$ " Ans.

6. Drum ratio $= 60$ in. $\div 42$ in. $= 1.42$.

1st expansion blades $= \dfrac{2.5 \times 1.4}{1.42 \times 1.42 \times 2} = .86$ in., say $\frac{7}{8}$ in.
2nd " $= .875 \times 1.4 = 1.22$ in., say $1\frac{1}{4}$ in.
3rd " $= 1.25 \times 1.4 =$ $1\frac{3}{4}$ "
4th " $= 1.75 \times 1.4 = 2.45$ " $2\frac{1}{2}$ "
5th " $= 2.5 \times 1.4 =$ $3\frac{1}{2}$ "
6th " $= 3.5 \times 1.4 = 4.9$ " 5 "
7th " $=$ 5 "
8th " $=$ 5 " Ans.

$$\text{Steam flow at 7th expansion} = \frac{5000 \times 14 \times 90}{2 \times 60 \times 60} = 875 \text{ cub. ft.}$$

Required clear area between blades $= 875 \div 350 = 2.5$ sq. ft.

$$\text{Annulus area} = \frac{(60 + 5) \times 3.1416 \times 5}{144} = 7 \text{ sq. ft.}$$

Then, annulus factor $= 7 \div 2.5 = 2.8$.

$$\text{Steam flow at 8th expansion} = \frac{5000 \times 14 \times 140}{2 \times 60 \times 60} = 1361 \text{ cub. ft.}$$

Required clear area $= 1361 \div 380 = 3.58$ sq. ft.
Annulus area $= 7$ sq. ft. as formerly.
Then, annulus factor $= 7 \div 3.58 = 1.95$. Ans.

7. $\dfrac{8000 \times 13.5 \times 2.78}{60 \times 60} = 83.4$ cub. ft. flow per sec.

Then, $\dfrac{2700}{\sqrt{156}} = 217$ ft. per sec. steam velocity at 1st expansion.

$\dfrac{83.4}{217} = .384$ ft. $=$ required clear exit area between blades.

and $\sqrt{\dfrac{.384 \times 3 + 8.295}{.7854}} = 3.467$.

$\dfrac{3.467 - 3.25}{2} = .108$ ft. blade height at 1st expansion.

Answers.
$.108 \times 12 = 1.29$ in., say $1\frac{1}{4}$ in. height of blades 1st expansion.
$1.25 \times 1.4 = 1.75$ " $1\frac{3}{4}$ " 2nd "
$1.75 \times 1.4 = 2.45$ " $2\frac{1}{2}$ " 3rd "
$2.5 \times 1.4 = 3.5$ " $3\frac{1}{2}$ " 4th "

8. Drum ratio $= 55 \div 39 = 1.41$.

Blade heights $= \dfrac{3.5 \times 1.4}{1.41 \times 1.41 \times 2} = 1.23$ in., say $1\frac{1}{4}$ in. at 1st L.P. expansion.

Then,

Answers.

$1.25 \times 1.4 = 1.75$	in., say	$1\frac{3}{4}$ in. at	2nd L.P. expansion.
$1.75 \times 1.4 = 2.45$	"	$2\frac{1}{2}$ "	3rd "
$2.5 \times 1.4 = 3.5$	"	$3\frac{1}{2}$ "	4th "
$3.5 \times 1.4 = 4.9$	"	5 "	5th "
$5 \times 1.4 =$	"	7 "	6th "
"Wing" blades		7 "	7th "
		7 "	8th "

9. H.P.

$$\frac{10000 \times 13.2 \times 2.78}{60 \times 60} = 101.93 \text{ cub. ft. flow per sec.}$$

$$\frac{2700}{\sqrt{144}} = 225 \text{ ft. per sec. steam velocity.}$$

$101.93 \div 225 = .453$ ft. = exit area between blades.

Then, $\sqrt{\dfrac{11.0446 + .453 \times 3}{.7854}} = 3.97$ ft. diam. across blade tips.

$\therefore \dfrac{3.97 - 3.75}{2} = .11$ ft. blade height at 1st H.P. expansion.

Answers.

$\therefore .11 \times 12 = 1.32$	in., say	$1\frac{3}{8}$ in. height at	1st expansion.
$1.375 \times 1.4 = 1.92$	"	2 "	2nd "
$2 \times 1.4 = 2.8$	"	$2\frac{7}{8}$ "	3rd "
$2.875 \times 1.4 =$		4 "	4th "

10. Drum ratio $= 64 \div 45 = 1.42$.

Then, $\dfrac{4 \times 1.4}{1.42 \times 1.42 \times 2} = 1.39$ in., say $1\frac{3}{8}$ in. blade height at 1st expansion.

And

Answers.

$1.375 \times 1.4 = 1.92$	in., say	2 in. blade height at	2nd expansion.
$2 \times 1.4 = 2.8$	"	$2\frac{7}{8}$ "	3rd "
$2.875 \times 1.4 =$		4 "	4th "
$4 \times 1.4 = 5.6$	"	$5\frac{1}{2}$ "	5th "
$5.5 \times 1.4 = 7.7$	"	$7\frac{3}{4}$ "	6th "
7.7	"	$7\frac{3}{4}$ "	7th "
7.7	"	$7\frac{3}{4}$ "	8th "

11. $\dfrac{14000 \times 13.4 \times 2.96}{60 \times 60} = 151.47$ cub. ft. flow per sec.

$2800 \div 11.83 = 236.68$ ft. per sec. steam velocity.

$\therefore 151.47 \div 236.68 = .639$ ft. exit area between blades.

Then, $\sqrt{\dfrac{15.904 + .639 \times 12}{.7854}} = 4.76$ diameter across blade tips.

$\therefore \dfrac{4.76 - 4.5}{2} \times 12 = 1.56$ in., say $1\frac{1}{2}$ in. blade height at 1st expansion.

Ans.

$\therefore 1.5 \times 1.42 = 2.13$	in., say	$2\frac{1}{4}$ in. blade height at	2nd expansion.
$2.25 \times 1.42 = 3.19$	"	$3\frac{1}{4}$ "	3rd "
$3.25 \times 1.42 = 4.61$	"	$4\frac{5}{8}$ "	4th "

12. Drum ratio $= 78 \div 54 = 1.44$.

Then, $\dfrac{4.625 \times 1.42}{1.44 \times 1.44 \times 2} = 1.58$ in., say $1\frac{5}{8}$ in. blade height at 1st expansion.

Answer.

∴ $1.625 \times 1.42 = 2.30$	in., say $2\frac{3}{8}$	in. blade height at	2nd	expansion.
$2.375 \times 1.42 = 3.37$	" $3\frac{3}{8}$	"	3rd	"
$3.375 \times 1.42 = 4.79$	" $4\frac{3}{4}$	"	4th	"
$4.75 \times 1.42 = 6.74$	" $6\frac{3}{4}$	"	5th	"
$6.75 \times 1.42 = 9.58$	" $9\frac{1}{2}$	"	6th	"
	$9\frac{1}{2}$	"	7th	"
	$9\frac{1}{2}$	"	8th	,

13. $\dfrac{5500 \times 14 \times 2.31}{60 \times 60} = 49.40$ cub. ft. flow per sec.

Then, $\dfrac{3000}{\sqrt{121}} = \dfrac{3000}{11} = 272.72$ ft. per sec. velocity.

∴ $49.40 \div 272.72 = .181$ ft.,

and $.181 \times 3 = 543$ sq. ft. exit area.

∴ $\sqrt{\dfrac{10.559 + .543}{.7854}} = 3.75$ ft.,

and $\dfrac{3.75 - 3.66}{2} = .045$ ft.

Answers.

∴ $0.45 \times 12 = .540$	in., say $\frac{1}{2}$	in. blade height at	1st	expansion.
$.5 \times 1.4 = .70$	" $\frac{3}{4}$	"	2nd	"
$.75 \times 1.4 = 1.05$	" 1	"	3rd	"
$1 \times 1.4 = 1.4$	" $1\frac{1}{2}$	"	4th	"

14. Drum ratio $= 66 \div 44 = 1.5$.

Then, $\dfrac{1.5 \times 1.4}{1.5 \times 1.5 \times 2} = .46$ in., say $\frac{1}{2}$ in. height of blades at 1st expansion.

Ans.

∴ $.5 \times 1.4$	$= .70$	in., say $\frac{3}{4}$	in. height of blades at	2nd	expansion.
$.75 \times 1.4$	$= 1.05$	" 1	"	3rd	"
1×1.4	$= 1.4$	" $1\frac{1}{2}$	"	4th	"
1.5×1.4	$= 2.10$	" $2\frac{1}{8}$	"	5th	"
2.125×1.4	$= 2.975$	" 3	"	6th	"
	$= 2.975$	" 3	"	7th	"
	$= 2.975$	" 3	"	8th	"

15. $\dfrac{11000 \times 13.8 \times 2.26}{60 \times 60} = 95.290$ cub. ft. flow per sec.

$\dfrac{3000}{\sqrt{144}} = 250$ ft. per sec. velocity.

∴ $95.290 \div 250 = .381$,

and $.381 \times 3 = 1.143$ ft. exit area.

∴ $\sqrt{\dfrac{23.758 + 1.143}{.7854}} = 5.63$ ft.

$\therefore \dfrac{5.63 - 5.5}{2} = .065$ ft. blade height at 1st expansion.

Answers.	And $.065 \times 12 = .780$ in., say	$\frac{7}{8}$ in.	blade height	1st expansion.		
	$.875 \times 1.4 = 1.22$	„	$1\frac{1}{4}$	„	2nd	„
	$1.25 \times 1.4 =$		$1\frac{3}{4}$	„	3rd	„
	$1.75 \times 1.4 = 2.45$	„	$2\frac{1}{2}$	„	4th	„
	$2.5 \times 1.4 =$		$3\frac{1}{2}$	„	5th	„
	$3.5 \times 1.4 = 4.9$	„	5	„	6th	„

16. Drum ratio $= 89 \div 66 = 1.348$.

Then,

$\dfrac{5 \times 1.4}{1.348 \times 1.348 \times 1} = 3.85$ in., say $3\frac{7}{8}$ in. blade height at 1st expansion.

Answers.	$\therefore\ 3.875 \times 1.4 = 5.42$ in., say	$5\frac{1}{2}$ in.	blade height at	2nd expansion.		
	$5.5 \times 1.4 = 7.70$	„	$7\frac{3}{4}$	„	3rd	„
	$7.75 \times 1.4 = 10.85$	„	11	„	4th	„
			11	„	5th	„
			11	„	6th	„

17. $\dfrac{8300 \times 11.5 \times 46}{60 \times 60} = 1219.63$ cub. ft. flow per sec.

Then, $1219.63 \div 600 = 2.032$ sq. ft. exit area.

$\therefore\ 2.032 \times 3 = 6.096$ total area.

Then, $\sqrt{\dfrac{44.17 + 6.096}{.7854}} = 8.$

$\dfrac{8 - 7.5}{2} = .25$ ft. blade height at 1st expansion.

Answers.	$.25 \times 12 =$		3 in.	height of blades at	1st expansion.	
	$3 \times 1.33 = 3.99$	in., say	4	„	2nd	„
	$4 \times 1.33 = 5.32$	„	$5\frac{1}{2}$	„	3rd	„
	$5.5 \times 1.33 = 7.31$	„	$7\frac{1}{2}$	„	4th	„
	$7.5 \times 1.33 = 9.97$	„	10	„	5th	„
			10	„	6th	„

18. $\dfrac{3000 \times 13 \times 41}{60 \times 60} = 444.166$ cub. ft. flow per sec.

$444.166 \div 400 = 1.110.$

$\therefore\ 1.110 \times 3 = 3.330$ exit area.

Then, $\sqrt{\dfrac{28.274 + 3.330}{.7854}} = 6.34$ diameter across blade tips.

$\therefore\ \dfrac{6.34 - 6}{2} = .17$ ft. blade height at 1st expansion.

Answers.	$.17 \times 12 = 2.04$ in., say	$2\frac{1}{8}$ in.	blade height at	1st expansion.		
	$2.125 \times 1.4 = 2.97$	„	3	„	2nd	„
	$3 \times 1.4 = 4.2$	„	$4\frac{1}{4}$	„	3rd	„
	$4.25 \times 1.4 = 5.9$	„	6	„	4th	„
			6	„	5th	„

Turbine Shaft Horse-Power.

1. H.P. turbine horse-power $\frac{1.53 \times .68 \times 6.75^4 \times 600}{12.5 \times 36} = 2879$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .68 \times 6.75^4 \times 650}{12.5 \times 36} = 3119$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .68 \times 6.75^4 \times 654}{12.5 \times 36} = 3138$ horse-power.

Total horse-power $= 2879 + 3119 + 3138 = 9136$.

$\therefore$ Equivalent horse-power $= 9136 \div .9 = 10151$ horse power. Answer.

2. H.P. turbine horse-power $\frac{1.53 \times .8 \times 4.75^4 \times 690}{12.5 \times 33} = 1042$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .8 \times 4.75^4 \times 830}{12.5 \times 33} = 1253$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .8 \times 4.75 \times 825}{12.5 \times 33} = 1246$ horse-power.

Total horse-power $= 1042 + 1253 + 1246 = 3541$ horse-power.

$\therefore$ Equivalent horse-power $= 3541 \div .9 = 3934.4$ horse-power. Answer.

3. H.P. turbine horse-power $\frac{1.53 \times .35 \times 6^4 \times 490}{12.5 \times 25} = 1088$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .35 \times 6^4 \times 580}{12.5 \times 25} = 1288$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .35 \times 6^4 \times 582}{12.5 \times 25} = 1292$ horse-power.

Total horse-power $= 1088 + 1288 + 1292 = 3668$ horse-power.

$\therefore$ Equivalent horse-power $= 3668 \div .9 = 4075.5$ horse-power. Answer.

4. H.P. turbine horse-power $\frac{1.53 \times .6 \times 10^4 \times 300}{12.5 \times 60} = 3672$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .6 \times 10^4 \times 300}{12.5 \times 60} = 3672$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .6 \times 10^4 \times 300}{12.5 \times 60} = 3672$ horse-power.

Total horse-power $= 3672 + 3672 + 3672 = 11016$.

$\therefore$ Equivalent horse-power $= 11016 \div .9 = 12240$ horse-power. Answer.

5. H.P. turbine horse-power $\frac{1.53 \times .358 \times 8^4 \times 625}{12.5 \times 40} = 2840$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .358 \times 8^4 \times 630}{12.5 \times 40} = 2826$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .358 \times 8^4 \times 632}{12.5 \times 40} = 2835$ horse-power.

Total horse-power $= 2840 + 2826 + 2835 = 8465$ horse-power.

$\therefore$ Equivalent horse-power $= 8465 \div .9 = 9405.5$ horse-power. Answer.

6. H.P. turbine horse-power $\frac{1.53 \times .3 \times 11^4 \times 230}{12.5 \times 40} = 3091.3$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .3 \times 11^4 \times 230}{12.5 \times 40} = 3091.3$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .3 \times 11^4 \times 230}{12.5 \times 40} = 3091.3$ horse-power.

Total horse-power $= 3091.3 + 3091.3 + 3091.3 = 9273.9$ horse-power.

$\therefore$ Equivalent horse-power $= 9273.9 \div .9 = 10304.3$ horse-power. Answer.

7. H.P. turbine horse-power $\frac{1.53 \times 1 \times 14.5^4 \times 190}{18.5 \times 100} = 6946$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times 1 \times 14.5^4 \times 190}{18.5 \times 100} = 6946$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times 1 \times 14.5^4 \times 190}{18.5 \times 100} = 6946$ horse-power.

Total horse-power $= 6946 + 6946 + 6946 = 20838$ horse-power.

$\therefore$ Equivalent horse-power $= 20838 \div .9 = 23153$ horse-power. Answer.

8. S.H.P. turbine horse-power $\frac{1.53 \times .438 \times 22^4 \times 160}{18.5 \times 80} = 16971$ horse-power.

S.L.P. turbine horse-power $\frac{1.53 \times .438 \times 22^4 \times 160}{18.5 \times 80} = 16971$ horse-power.

P.H.P. turbine horse-power $\frac{1.53 \times .438 \times 22^4 \times 160}{18.5 \times 80} = 16971$ horse-power.

P.L.P. turbine horse-power $\frac{1.53 \times .438 \times 22^4 \times 160}{18.5 \times 80} = 16971$ horse-power.

Total horse-power $= 16971 + 16971 + 16971 + 16971 = 67884$ horse-power.

$\therefore$ Equivalent horse-power $= 67884 \div .9 = 75426$ horse-power. Answer.

Horse-Power Developed in Blade Rows.

1. Horse-power per blade row $\Big\} = \frac{(322^2 - 217^2) \times .847 \times 22.5 \times 60}{64.4 \times 33000} = 30.45$ horse-power.

Horse-power per expansion $\Big\} = 30.45 \times 8 = 243.6$ horse-power.

Heat drop per expansion $\Big\} = 243.6 \times 33000 \div 778 = 10332.6$ B.T.U.

Ratio V_t to V_{s3} $= \left(\frac{3.7083 \times 3.1416 \times 650}{60}\right) \div 322 = .39$ ratio.

Ans.

2. Horse-power per blade row $\Big\} = \frac{(360^2 - 245^2) \times .83 \times 22.5 \times 60}{64.4 \times 33000} = 36.68$ horse-power.

Horse-power per expansion $\Big\} = 36.68 \times 8 = 293.44$ horse-power.

Heat drop per expansion $\Big\} = 293.44 \times 33000 \div 778 = 12446.6$ B.T.U.

Ratio V_t to V_{s3} $= \left(\frac{3.7708 \times 3.1416 \times 650}{60}\right) \div 360 = .356$ ratio.

Ans.

3. Horse-power per blade row $= \dfrac{(487^2 - 350^2) \times .82 \times 11.25 \times 60}{64.4 \times 33000} = 29.86$ horse-power.
Horse-power per expansion $= 29.86 \times 8 = 238.88$ horse-power.
Heat drop per expansion $= 238.88 \times 33000 \div 778 = 10132.44$ B.T.U.
Ratio V_t to V_{s3} $= \left(\dfrac{4.4375 \times 3.1416 \times 650}{60}\right) \div 487 = .310$ ratio.
Ans.

4. Horse-power per blade row $= \dfrac{(541^2 - 397^2) \times .76 \times 11.25 \times 60}{64.4 \times 33000} = 32.6$ horse-power.
Horse-power per expansion $= 32.6 \times 8 = 260.8$ horse-power.
Heat drop per expansion $= 260.8 \times 33000 \div 778 = 11062.2$ B.T.U.
Ratio V_t to V_{s3} $= \left(\dfrac{4.625 \times 3.1416 \times 650}{60}\right) \div 541 = .29$ ratio.
Ans.

5. Horse-power per blade row $= \dfrac{(600^2 - 475^2) \times .74 \times 11.5 \times 60}{64.4 \times 33000} = 31.58$ horse-power.
Horse-power per expansion $= 31.58 \times 8 = 252.64$ horse-power.
Heat drop per expansion $= 252.64 \times 33000 \div 778 = 10716.09$ B.T.U.
Ratio V_t to V_{s3} $= \left(\dfrac{4.66 \times 3.1416 \times 650}{60}\right) \div 600 = .264$ ratio.
Ans.

6. Horse-power per blade row $= \dfrac{(750^2 - 646^2) \times .715 \times 11.25 \times 60}{64.4 \times 33000} = 32.97$ horse-power.
Horse-power per expansion $= 32.97 \times 8 = 263.76$ horse-power.
Heat drop per expansion $= 263.76 \times 33000 \div 778 = 11187.7$ B.T.U.
Ratio V_t to V_{s3} $= \left(\dfrac{4.75 \times 3.1416 \times 650}{60}\right) \div 750 = .215$ ratio.
Ans.

Steam Condensed in Turbines.

1. $112000 \times (1 - .85) = 112000 \times .15 = 16800$ lbs. Answer.

2. $6000 \times 13.5 = 81000$ lbs. steam.
$\therefore\ 81000 \times (1 - .848) = 81000 \times .152 = 12312$ lbs. per hour. Answer.

3. $45000 \times \frac{13}{2} = 292500$ lbs. steam.
$\therefore\ 292500 \times (1 - .835) = 292500 \times .165 = 48262.5$ lbs.
Then $48262.5 \div 2240 = 21.54$ tons. Answer.

4. $$\frac{10000 \times 14}{60} = 2333.3 \text{ lbs. steam per minute.}$$

$$\therefore\ 2333.3 \times (1 - .932) = 2333.3 \times .068 = 158.6 \text{ lbs. per minute.}$$

Answer.

5. $$\frac{10000 \times 14}{60} = 2333.3 \text{ lbs. per minute.}$$

$$\therefore\ 2333.3 \times (1 - .85) = 2333.3 \times .15 = 349.9, \text{ say } 350 \text{ lbs. per minute.}$$

Answer.

6. $$68000 \times 13.7 = 931600 \text{ lbs. steam.}$$

$$\therefore\ 931600 \times (1 - .848) = 931600 \times .152 = 141603.2 \text{ lbs.}$$

And, $141603.2 \div 2240 = 63.21$ tons. Answer.

APPENDIX.

GENERAL NOTES.

Efficiency of "Lusitania" Turbines.—Referring to the paper read by Thomas Bell, Esq. (page 153), on the trial and sea performances of the "Lusitania," the steam consumption of the turbines at an average shaft horse-power of 65,000 is given as 13.1 lbs. per hour. Assuming an H.P. initial gauge pressure of 150 lbs., and a terminal L.P. pressure of 1.5 lbs. absolute, we can determine the over-all turbine efficiency as follows :—

Take adiabatic expansion, with H.P. initial steam of .99 dryness, and L.P. terminal steam of .77 dryness.

Then, $150 + 15 = 165$ lbs. absolute, temperature 366 deg.,
and, $366 + 461 = 827$ absolute temperature,
165 lbs. absolute = 856.2 B.T.U. latent heat,
1.5 ,, = 115.9 temperature,
and $115.9 + 461 = 576.9$ absolute temperature,
1.5 lbs. absolute = 1033.2 B.T.U. latent heat.

$$\begin{aligned}\text{Heat drop} &= 856.2 \times .99 - 1033.2 \times .77 + 827 - 576.9\\ &= 847.638 - 795.564 + 827 - 576.9\\ &= 1372.46 - 1674.63\\ &= 302.17 \text{ B.T.U.}\end{aligned}$$

$$\text{Efficiency} = \frac{33000 \times 60}{\text{lbs. steam} \times \text{B.T.U.} \times 778} = \frac{33000 \times 60}{13.1 \times 302.17 \times 778} = .64.$$

The turbine efficiency is therefore equal to 64 per cent.

Again, it may be pointed out that as the steam consumption is given per *shaft* horse-power per hour the consumption will be still less per I.H.P., and taking the ratio of B.H.P. to I.H.P. as .93 is to 1 for this special case of large power, then, $13.1 \times .93 = 12.18$ lbs. steam per I.H.P. per hour for turbines alone. The total steam consumption (including all auxiliary gear) is given as 15.35 lbs. per shaft horse-power per hour, and again reducing to the I.H.P. standard we get $15.35 \times .93 = 14.27$ lbs. steam per I.H.P. per hour, which may be considered as a very satisfactory result indeed, and one comparing most favourably with the best modern marine reciprocating engine practice.

Dummy Readings (Actual Practice).—The following readings of the micrometer gauge show the dummy clearance under three conditions:—

	Port.	H.P.	Starboard.
Taken when cold - -	.018	.026	.025
Taken when heated up - -	.037	.030	.034
Taken when running - - -	.021	.021	.025

It will be noticed that the clearance is at a maximum in each turbine when "heated up," and afterwards falls away when the turbines are running. This is probably due to the propeller thrust forcing the shaft forwards and so slightly reducing the clearance as measured.

Wear Down of Main Bearings.—The following readings, taken by feelers, show the actual wear down in the main bearings of a cross-channel steamer after ten months' regular service:—

Turbine.	Forward.	Aft.
H.P. - -	0	0
P.L.P. - -	0	.003
S.L.P. - -	.004	.004

The maximum wear down was $\frac{4}{1000}$ of an inch in the case of the starboard L.P. turbine, the H.P. and port L.P. turbine forward end showing no wear down whatever.

Propulsive Efficiency.—Of the total I.H.P. developed by the engines only about 50 per cent. or thereabout is applied in the effective advance of the steamer when the various losses are eliminated—that is, the actual hull resistance in lbs. at any given speed, multiplied by the advance of the hull per minute and divided by the constant 33,000 foot-pounds, will give the effective horse-power, or, as usually expressed, the E.H.P. Therefore, E.H.P. ÷ I.H.P. = Propulsive efficiency. This efficiency can only be accurately determined by model tank experiments of resistance, after which the data so obtained is converted into terms of the actual hull by a series of calculations known as the "law of comparison," and devised by the late Dr Froude. The tank

experiments with the reduced scale hull models obtain progressive "tow rope" resistances which are the actual resistances at various speeds of the model hull (the propeller being omitted). These are made up as follows:—

Resistance.—1. Skin frictional resistance of the hull surface. 2. Wave making resistance of the hull body. 3. Eddy making resistance of the hull body.

The foregoing constitute what may perhaps be termed the true resistances, and to overcome these the effective horse-power is required.

Power Losses.—The losses of engine power are made up as follows:—

1. Friction (initial and load).
2. Propeller inefficiency.
3. Hull inefficiency.

The frictional losses are those occasioned by the working parts and the power absorbed by the thrust block. The propeller losses are due to excessive slip, blades friction, and other causes, and the hull efficiency is a result which may either be under or above unity, according to the difference between what is called "augmentation of resistance," due to the propeller blades at the stern, and "wake speed gain." Generally, however, the "wake speed gain" balances the augment of resistance to within a very few per cent., although an allowance of about 95 per cent. is often taken as the "hull efficiency."

The "wake speed" is produced by the water closing in on the stern as the hull advances, and this body of water acquires a forward motion or speed varying in degree with the lines of the hull body.

Utilisation of Power.—The total I.H.P. developed by the engines is therefore used up somewhat as follows, although it must be understood that the values given vary in different cases and under different conditions in the same case:—

Taking the total I.H.P. as 100 per cent.

Reciprocating Engines.

Indicated horse-power - - - - - - - -	100	per cent.
Engine friction loss - - - - - - - -	10	"
Horse-power at propeller - - - - - -	90	"
Propeller efficiency, 62 per cent.		
Then, $90 \times .62 = 55.8$ "		
Horse-power by propeller - - - - - -	55.8	"
Hull efficiency, 95 per cent.		
Then, $55.8 \times .95 = 53$ "		
Effective horse-power - - - - - - - -	53	"

Therefore, propulsive efficiency $= \frac{53}{100}$, or $.53$.

Turbine Engines.

Shaft horse-power - - - - - - - - 100 per cent.
Propeller efficiency, 60 per cent.
Then, $100 \times .60 = 60$ „
Horse-power by propeller - - - - - - 60 „
Hull efficiency, 95 per cent.
Then, $60 \times .95 = 57$ „
Effective horse-power - - - - - - - 57 „

Therefore, propulsive efficiency $= \frac{57}{100}$, or .57.

NOTE.—It should be observed that relatively the propulsive efficiency appears higher when referred to B.H.P. than to I.H.P. as in the foregoing case.

Referring to the propulsive efficiency of the U.S. cruiser scout "Salem," (page 254), the figures given, 62 per cent., are based on the B.H.P., and reducing this to the I.H.P. standard by allowing a 10 per cent. engine friction loss, then, $62 \times .90$ gives 55.8 as the actual propulsive efficiency, a result much above the average in turbine steamers.

Power Curves and Propulsive Efficiency.—If a curve is plotted out from the tank experiments giving the required or effective horse-power at various speeds, and above this another curve showing the actual I.H.P. developed during the progressive speed trials, then by dividing the E.H.P. by the I.H.P. on the same vertical line the propulsive efficiency is obtained.

This efficiency should be highest at the maximum speed and power.

It will thus be obvious that for a given hull and speed the effective horse-power is a constant quantity, but the I.H.P. for the same may be lower or higher according to the efficiency of the line shafting, propeller and hull, as, assuming an ideal case without losses of any kind, the E.H.P. would be equal to the I.H.P. and the propulsive efficiency equal to 100 per cent., or 1.

Loss of Power due to Windage or Blade Resistance in Idle Turbines.—No turbine can be truly said to revolve "idly" in a vacuum, as in no case is the vacuum absolute, and with the usual vacuum carried in the reverse turbines (when incorporated with the ahead) of, say, from 22 in. to 26 in., the air resistance varies from 4 lbs. to 2 lbs. absolute; this in itself is quite sufficient to produce a considerable loss of power due to windage or blade resistance, which loss has, until quite recently, not been appreciated at its full value. The negative work produced in this way cannot be accurately determined by calculation, and experiment only can give an approximate idea of the amount. There is no doubt that the importance of a high degree of vacuum (when running full power with the main turbines or in the ahead turbines when running astern) both in the condenser and in the idle turbines, such as the reverse and cruising turbines, cannot be over-estimated. This windage accounts chiefly for the higher steam consumption experienced when running astern for any lengthened period. Again, in the case of independent astern turbines, the loss of power due to blade resistance is increased, as the vacuum is then not

so good as when the reverse turbines are incorporated. This might be overcome by providing a suction pipe to the dry air pump from the reverse turbines, or from the cruising turbines, to be open when running ahead full power, which arrangement would, however, have the disadvantage of complicating the connections, and would also act as a hindrance to quick reversing of the turbines. Turbine over-all efficiency would undoubtedly be much higher if a vacuum of, say, from 28 in. or 29 in. could be obtained in the "idle" turbines.

Condition of Turbines when opened up after Service.—If the boilers all keep in good working order and priming prevented, the turbines, after even a year's hard running, do not as a rule show any evidence of corrosion or wear, and in a case noted recently by the writer, the turbines were in absolutely perfect condition, no marks of any kind being visible on the dummies, casings, rotors, or blading. To obtain this desirable result, the boilers must be strictly attended to, and all tendency to priming checked at once. The density in the case referred to above did not exceed a maximum of 2 ounces per gallon.

If the reverse turbines are often in use (as in the case of channel steamers), the astern side of the thrust block rings are found to wear much more than the ahead; this indicates that the propeller thrust when running astern is in excess of the steam thrust on the reverse turbine blades. The astern wear can be taken up by screwing the top half of the thrust cover forward by means of the adjusting stud and gear. (See page 107.)

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